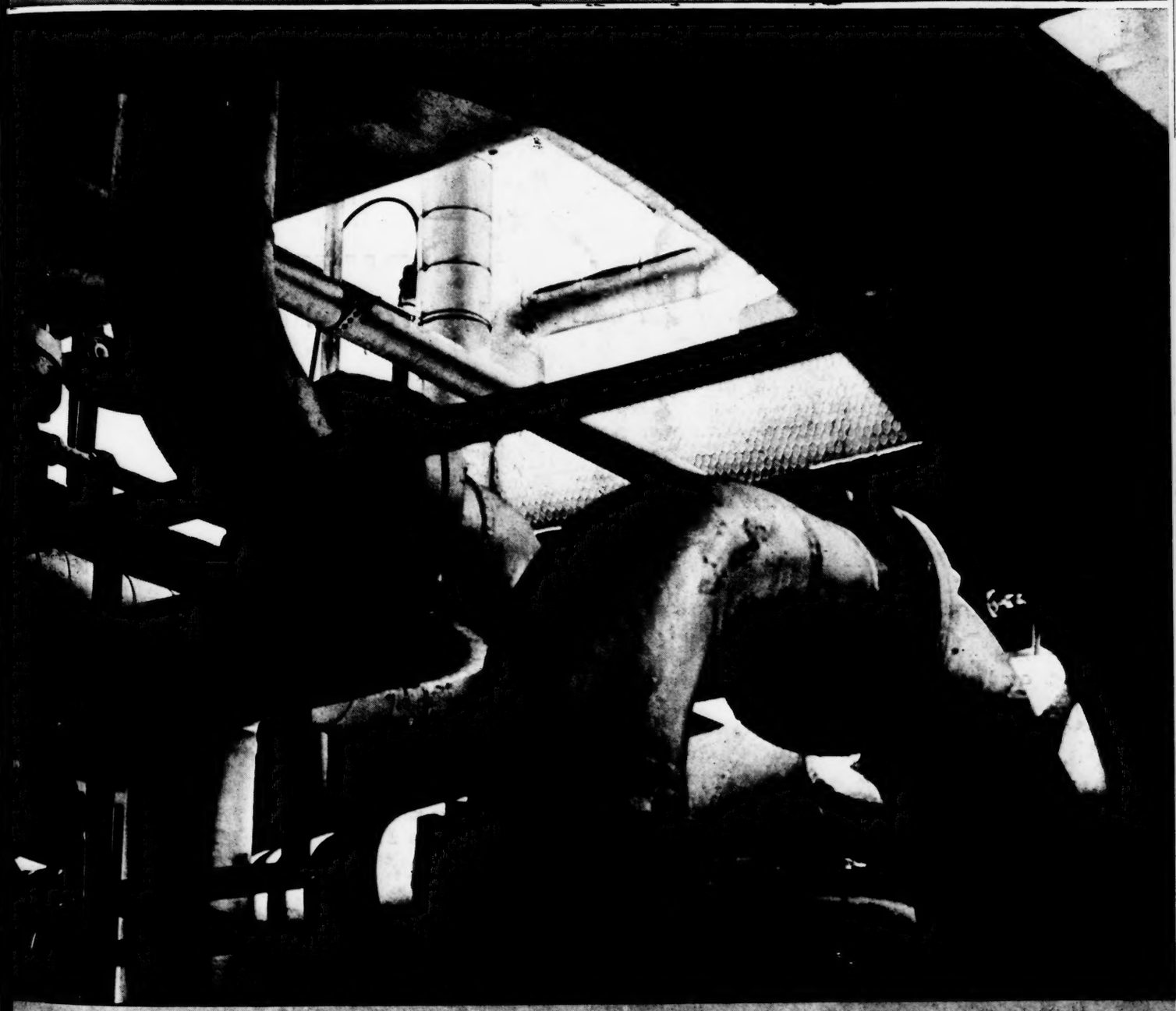
HEATING AND VENTILATING

AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION



The solvent recovery system at Shell's Wood River Refinery is a pipe joiner's dream . . . page 99

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Heat Transfer with Finned Tubes . page 83

Pre-Insulated

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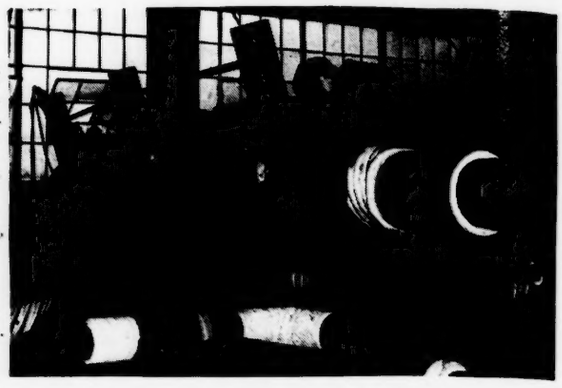
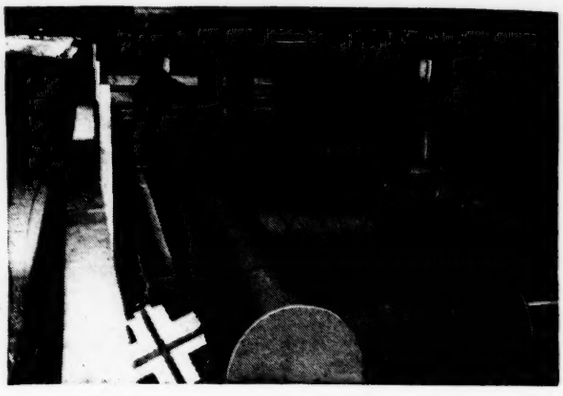


Units are completely fabricated and insulated at the Ric-wiL plant . . .

and are shipped ready to install with single-handling upon arrival.



by RIC-WIL *FOR OUTDOOR USE* Underground or Overhead

● An exclusive, outstanding feature of Ric-wiL Pipe Units is the fact that they are not only prefabricated but are also completely *insulated* in the Ric-wiL plant and are shipped ready for immediate, *single-handling* installation, per plans and specifications.

Ric-wiL Foilclad Units, with pipe and insulation coated with asphalt and wrapped with aluminum or copper foil, are supplied for overhead single-pipe lines. For underground lines or overhead multiple pipe systems, or wherever high strength conduit is required, Ric-wiL supplies units in which piping and insulation are housed in Helcor conduit. This helically corrugated galvanized iron conduit is full welded and pressure tight and covered with a heavy coating of asphalt and a wrapping of asphalt saturated asbestos for complete protection.

Either type unit is furnished complete with expansion loops, fittings, etc., in the same design, ready for field installation in minimum time and at minimum cost. In multiple pipe units, pipes may be insulated from each other or from the exterior only, as in the type of unit used for transmission of oil and process liquids. No matter how complicated the requirements of an individual job, Ric-wiL is equipped to prefabricate and insulate in the factory, to meet individual plans and specifications.

The insulated pipe headers shown above are typical examples. Designed for a system of piping actified and foul solutions, they required special insulating and cutting, mitering and welding of pipe with flanges and conduit. One 28 ft. unit comprises 3 tee branches, 1 Y branch, 2 reducers and 4 slip-on flanges welded to the pipe. Only work required at job-site was simple connection of header to adjacent equipment.

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VENTILATING

A MONTHLY MAGAZINE FOR ENGINEERS AND CONTRACTORS CONCERNED WITH THE DESIGN AND APPLICATION OF AIR CONDITIONING, REFRIGERATION, PIPING, AIR SANITATION, HEATING AND VENTILATION IN BUILDINGS

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PUBLISHERS OF MACHINERY & MACHINERY'S HANDBOOK
The INDUSTRIAL PRESS

THIS MONTH'S COVER

Close-up of pipes in the Duo-Sol extraction plant at Shell Oil Co.'s, Wood River, Ill., refinery. The intake of this plant is approximately 3,000 barrels a day. Note flanged, welded, and screwed pipe joints discussed in the series of articles continued this month with an analysis of welded and brazed joints beginning page 99.

NO BEARINGS IN THE AIR STREAM *of the*

Here's the answer to real economy in your air handling proble[illegible]e ELBOW AXIFLO, an angular-inlet type fan that actually has no bearing[illegible]r or belts in its air stream! This means *low maintenance* because bearings[illegible]cessible, and are unaffected by heat, dust or fumes. This important feature m[illegible]XIFLO applicable for handling hot, dust-laden gasses and for many other [illegible]t jobs.

ELBOW AXIFLO Fan offers pressure characteristics comparable to [illegible] centrifugal fans, particularly at the lower pressures where it is not usua[illegible]pply centrifugal fans at their peak efficiency. Yet AXIFLO is smaller, weighs less, takes less effort and expense to install and operate.

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ELBOW AXIFLO can be furnished with fan cooled shaft and water cooled bearings as optional equipment for 250° to 900° F. applications.

ELBOW AXIFLO can be installed with inlet-direction at practically any desired angle. Rigid angle-iron supports provide universal mounting.

Compact, efficient, versatile and economical—an ELBOW AXIFLO Fan is available NOW to fill *your* needs. Call your nearest Sturtevant office, or .....

Write Today!

*Trade-Mark of the Sturtevant Division Westinghouse Electric Corporation Reg. U.S. Pat. Off.

Washington News

LORING F. OVERMAN

IT seemed like old times in Washington during the closing days of October, with President Truman summoning Congress for an extra session; with resumption of certain allocations and price controls of scarce materials rumored, and with the word, "emergency" given a big play in every official utterance.

In the face of these and many other evidences that Washington is again to concern itself more directly with the functioning of the American business machine, it is difficult to realize that VE and VJ days are behind us—not ahead. Many in the H & V field will recall that the original post-war schedule had anticipated that by now things would be back to normal; Europe would have had two years in which to work out her destiny; the pipelines of domestic demand were to have been filled, and all of us could more or less relax on a business-as-usual scale.

What a rude awakening! Europe, China, and most of the rest of the world seem to be almost as bad off as when the bombs were falling; American production has made hardly a dent even in domestic demand, and the current peace is far from peaceful. National and international puzzlement over the turn of events seems the only certainty.

Ever since VJ-Day, many have wished for a return of that unity of purpose which enabled Government, Management and Labor to flood the world with production that won battles on a hundred scattered fronts. That need has apparently reached its zenith, and it is to be hoped that the coming special session of Congress may be instrumental in developing it. If unity of purpose can be demonstrated in Washington, perhaps the example will be noted elsewhere throughout the country—whether current bottlenecks be attributable to Management, or to Labor.

Best hope of tempering political tendencies with unifying facts is the current return of more than 100 Senators and Representatives—members of Congressional committees sent to Europe for the express purpose of finding out what's what, and why, and what can be done about it. If European conditions are as hazardous as is claimed, these men and women should be in a position to know it, and to pass the message on to colleagues who have remained in touch with situations on the home front. Possibly the combination of these two viewpoints will produce twin solutions for both overseas and homefront problems.

Inseparably Linked

Both, it appears, are closely linked, whether we like to admit it or not. Take for example, the October report of Secretary of the Interior Krug, on the ability of this country to live up to commitments contemplated in the Marshall plan for the aid of Europe. The Krug report states that the shortage of steel and steel products would be one of the principal bottlenecks in maintaining or increasing production.

Even with steel exports held to a level far below that contemplated by the Marshall plan, the steel shortage has handicaped manufacturers of heating and ventilating equipment, automobiles, machinery and others, including the oil industry, which needs more pipe lines, steel tubing and other equipment. Coal output, in turn, has been cut by shortages of open-top steel cars, and the shipping of wheat and other essentials has been traceable to inadequate steel supplies. In addition to such shortages as have been noted in this country, a "crippling shortage" is reported in foreign countries, where the United States is called upon to provide 50% of requirements, instead of the prewar 38%.

Capacity Not Enough

The Krug report observes that vast increases in productive capacity cannot solve the problem—at least not within the period of the present emergency—for the chief bottleneck to steel production is the shortage of raw materials, particularly steel scrap. Even the existing steel-making capacity, the report states, is not being fully used because of the lack of scrap, poor quality of some available coking coal, and high production costs generally.

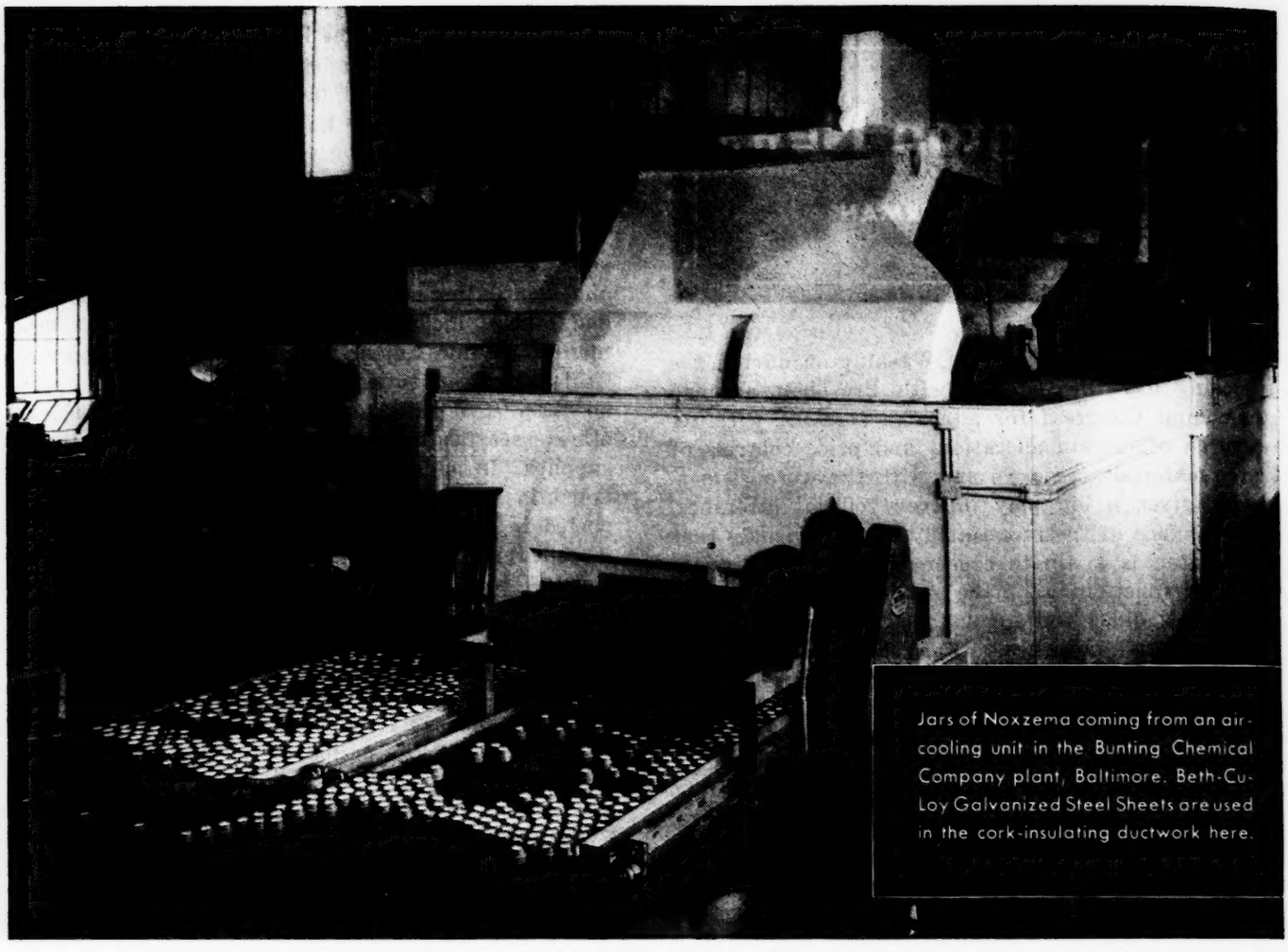
Relief of pressing shortages depends primarily, the report states, upon more effective utilization of the producing facilities at hand, as well as efforts to reduce foreign drains on our resources by aiding the efforts of other countries to increase the output of their own steel producing industries. To make the program of foreign aid yield maximum returns, the report concludes that Government and industry must also cooperate in controlling exports and channeling domestic supplies in order to insure their availability for essential needs at home.

What, Allocations?

It is at points like this that business individuality may be expected to collide with the Truman suggestion that the Congressional special session give some consideration to the channeling of scarce materials. On the one hand the Krug report appears as a cooly, logical analysis of the steel situation. The steel industry doubtless will be pleased that the Krug report concurs in the industry's claim that vast expansion of facilities will not provide an answer to the immediate problem. From here on, however, the paths will doubtless separate.

Within the heating and ventilating industries, for example, the natural question would be: "Are they going to take steel away from me and give it to someone else?"

Who is there who would envy a Senator or Congressman who receives from a dozen different industrial



Jars of Noxzema coming from an air-cooling unit in the Bunting Chemical Company plant, Baltimore. Beth-Cu-Loy Galvanized Steel Sheets are used in the cork-insulating ductwork here.

HOW *Air-Cooling* SAVES TIME IN HANDLING OF MEDICINAL PREPARATION

The Bunting Chemical Company of Baltimore use modern air-cooling methods to speed the continuous-line packaging of their products.

Formerly, jars of their medicinal preparations were too hot to handle when they left the filling machine. They had to cool for a full day before they could be labeled, capped and packed in boxes.

Now, these jars take a quick trip on a wire conveyor belt that runs through an air-cooling unit. Their temperature is reduced immediately, and as a result they can be shipped the same day. No time is lost, no extra handling is necessary.

This application of air-cooling is typical of the trend toward greater use of temperature and humidity control to improve processes and step up production in many lines of manufacturing.

Air-conditioning requires good ductwork and that, in turn, calls for galvanized steel sheets able to resist corrosion. Beth-Cu-Loy galvanized steel sheets are now used in many industrial air-conditioning installations throughout the nation. The copper-bearing steel base used in these sheets resists rust 2 to 2½ times as long as does ordinary carbon steel. On top of this, Beth-Cu-Loy sheets have a bright coating of Prime Western zinc that acts as a further defense against corrosion. Remember the name Beth-Cu-Loy whenever you figure on any type of air-conditioning installation for industrial purposes.

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GALVANIZED
Steel Sheets

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sources, telegrams reading, "Don't let them put me out of business." Yet during the war, for the common good of all, American industry and labor worked hand in hand on a program of first things first.

Output Up

Despite the handicaps of steel shortages and other production difficulties, some members of the heating and ventilating family came in for favorable mention in the Department of Commerce monthly index of production gains, released October 27. Covering production during the month of August, the report shows that 14 of 20 construction materials included in the index recorded gains, and that mechanical stokers were among six items that established monthly peaks for 1947. Stoker production reached 107.2% of the 1939 monthly average during August, a gain of 23.1% over July. Production during August of this year was, however, below that of the same month in 1946.

Warm air furnaces, while establishing no monthly peak, were being produced at the rate of 206.8% of the 1939 monthly average. Production during the preceding month was at the 159.8% level. Cast iron radiation dropped to 80.2% during August, from 85.7% the previous month. High point for the year was in May, when the index figure was 99.3%.

Other construction materials which established monthly peaks were lumber, hardwood flooring, brick, cement, and concrete reinforcing bars. All of these have been considered bottlenecks to construction, and their important gains should mark a welcome improvement in the supply picture. The lumber output for August was nearly 10% higher than the preceding month, and output of hardwood flooring was at a rate nearly two and one-half times that of August, 1946.

Housing Catches On

Department of Commerce reports of the improving conditions in the building materials field are reflected in evidence that the housing program is at last beginning to click. Starts during September were 88,000 —higher than in August. Estimates are that total starts for the year will be 825,000, and that by next Spring the monthly rate of building will be 10% higher than at present. Observors feel that, after getting away to a slow start, prospective purchasers either became reconciled to high prices or were forced into decision by rising rents.

Once started, it is predicted that the movement will continue until the building industry makes an appreciable dent in the backlog of needed homes. Slight price declines are anticipated as the building program continues, with most of the reduction coming out of the "contingency allowance" which most speculative builders have figured in to take care of delays, anticipated but unpredictable. With building supplies catching up with demand, it will be more difficult to justify such contingencies.

There are indications that the heating and ventilating industries must continue to look to housing as a major market for 1948, supplemented by commercial structures, but with industrial building as a poor third. It is anticipated that commercial building will keep step with new housing communities, but that factories and industrial buildings will be held in the blueprint stage pending whatever break accompanies solution of the emergency housing problem.

It may be assumed that calling of the special session of Congress will call a halt on the just-started 40-city housing hearings planned by a joint congressional subcommittee. At a hearing on October 21 at Pittsburgh, Pa., the committee heard a score of witnesses, and as many explanations of the low production and high prices of homes.

Spokesmen for the Pittsburgh Real Estate Board blamed "slowdowns" by union workers. Union denials were voiced by the president of the Pittsburgh Building and Construction Trades Council. To blame, he said, were inexperienced and non-union workmen, and the tendency of contractors to pirate labor by means of bonus offers.

Despite the evidences of disagreement, members of the committee learned from the secretary of the Home Builders' Association of Allegheny County that 7,627 starts had been made thus far in 1947, compared with 3,370 starts during all of 1946. Sixty-five per cent of current construction is for rental purposes, he reported:

Coal OK – Oil Short

Interior Secretary Krug, it appears, had no lack of assignments during October. In addition to his Resources for Foreign Aid Report, he issued a report on October 23 forecasting that there will be virtually no shortage of industrial coal this year, and that household consumers will feel only a limited pinch in several scattered areas. On October 21, however, he appealed to the petroleum industry for advice in efforts to prevent a threatened shortage.

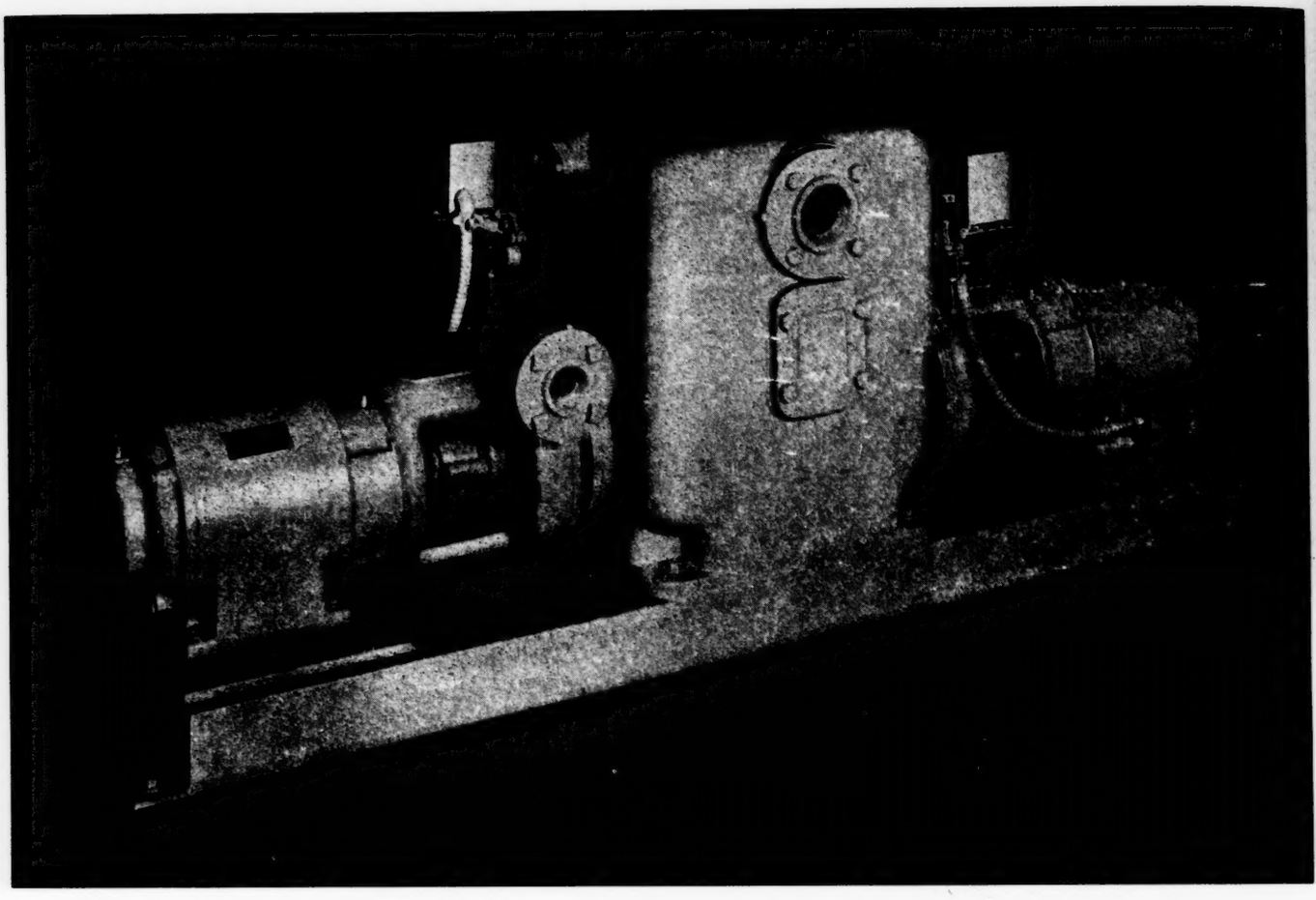
The favorable report on the coal outlook was presented to Secretary Krug by a 16-man advisory committee convened at his invitation from the ranks of the coal industry, organized labor, shippers, coal dealers, and the public. At the same time, the Bureau of Mines released a survey reporting coal stockpiles as of September 1 at 47,157,000 tons, representing a supply of 37 days. The Bureau of Mines statistics disclosed an estimated fiscal year production—by March 31, 1948—of 597,400,000 tons, a near-record for peacetime. Domestic and Canadian consumption is expected to require 563,000,000 tons, leaving 34,400,000 tons for export. Through September 25,200,000 tons had been shipped.

In his letter of appeal to the National Petroleum Council, Secretary Krug said the petroleum situation has deteriorated rapidly in recent weeks. He wrote:

"I consider the matter of sufficient gravity to call for immediate consideration by the council of such measures as can be taken by the Government, the industry and the public to conserve supplies of petroleum products or to increase their availability.

"There now appears to be little doubt that unless the winter is abnormally warm, shortages of petroleum products of varying degrees of severity will be widespread during the next 12 months with resultant physical and economic hardship to the public and to the industry."

MAXIMUM EFFICIENCY FOR STEAM HEATING SYSTEMS ON HEATING, DRYING AND PROCESS WORK.



JENNINGS CONDENSATION PUMPS

Designed for steam systems on heating, drying, or process work, these pumps offer the most reliable and efficient means of returning hot condensate to boiler or hot well. In many cases these pumps will save costly installation of boilers in a pit to provide gravity flow of returns from basement radiators and heating devices.

The Jennings Condensation Pumps are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Pumps are bronze fitted throughout, with tobin bronze shaft. The pumping impeller is of special design, made possible by our wide experience with return line heating pumps, and is especially adapted for handling hot water with the greatest possible efficiency.

Full automatic control is furnished by means of a ball float and float switch mechanism mounted on the receiving tank. All wiring is made up at the factory, and it is necessary only to connect the pump to the system and hook up the leads. Complete information regarding these economical pumps sent promptly on request.

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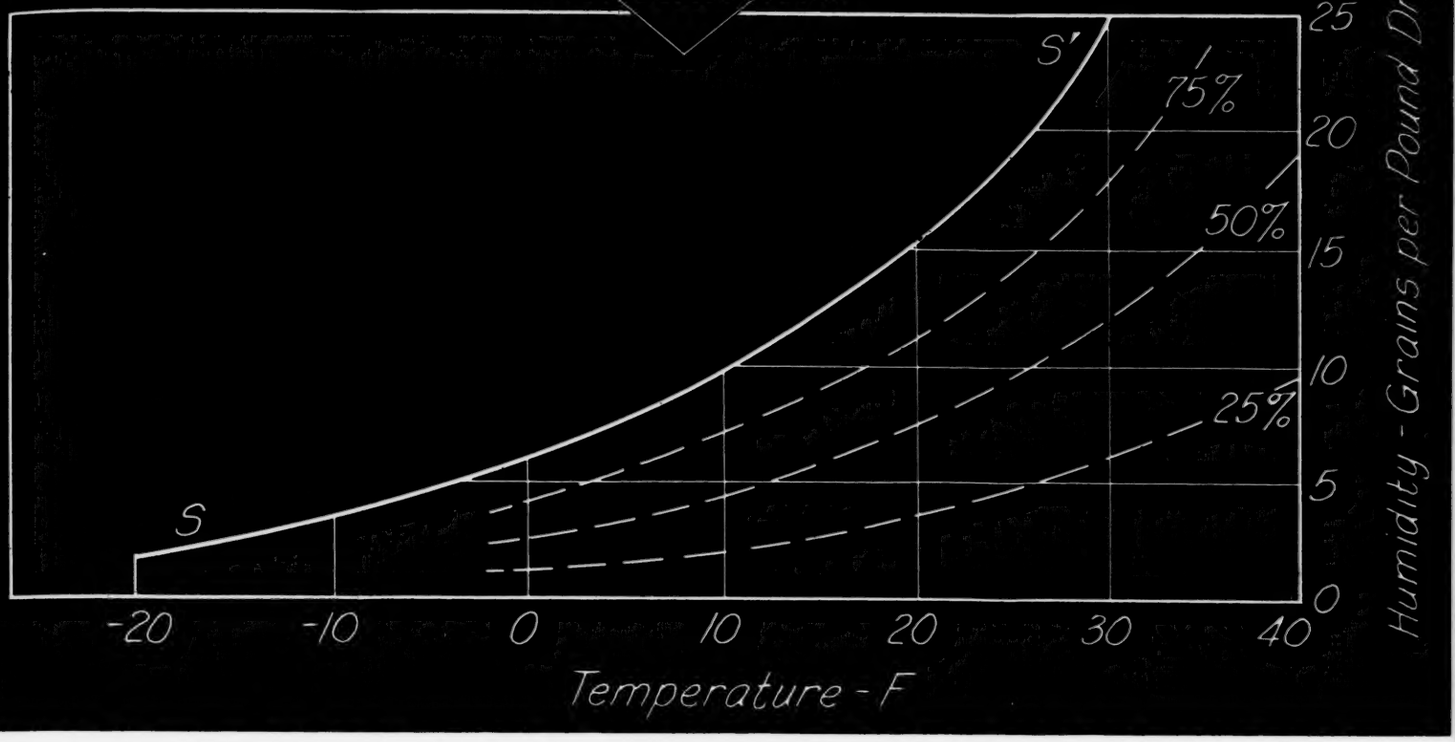

Part 1—Use of the Psychrometric Chart

For efficient chilling, freezing or storage processes, humidity as well as temperature must be taken into account. This article shows how both combine in the refrigeration cycle when plotted on the psychrometric chart. In this issue are presented Part 1—Use of the Psychrometric Chart, and Part 2—Condition of the Supply Air. Practical examples and conclusions will appear in future sections of this four-part article.

HUMIDITY is the most discussed and least understood of the factors governing refrigeration systems today. Many practical problems relate to humidity, yet aside from one or two rules of thumb, we have very limited knowledge of how to control it. Only recently, we have acquired the technical tools for better understanding, among which the psychrometric chart is the most important. Fig. 1 indicates how such a chart is put together, and it will be discussed first in this series of articles.

Better design for humidity control is greatly needed. Throughout most of the history of refrigeration, little attention has been paid to this matter. If goods desiccated, if excessive defrosting were required, it was just too bad.

The psychrometric chart provides a frame of reference to improve our thinking. It has been acquired from air conditioning, along with much of the know-how which is today available for making refrigeration a more exact science. This is curious. You might expect that air conditioning would have acquired its knowledge from the older art of refrigeration. But when the first chart was published by Mr. Carrier in 1911, it embodied nothing known to refrigeration, coming instead from the lore of the weather-man. The first commonly used low temperature charts have been in circulation only a very few years.

Construction of the Chart

This engineering creation might as well be called a "humidity chart". Its vertical scale is simply various amounts of water vapor or moisture in pounds or grains (one seven-thousandth of a pound), plotted against various temperatures as shown on the other scale. The reason why air conditioning was more interested in this kind of measurement was simply that people being cooled could tell you what they liked and disliked—and they are very sensitive to air conditions. An increase of humidity of a few percent is readily noticed, and this is especially striking when one considers that even when humidity stands at 100%, the water in the atmosphere amounts to less than 2% of the weight of the air present, at ordinary temperatures.

Now we find that stored goods are also sensitive. It is the intention of this and succeeding articles to show some practical conclusions as to the design and operation of refrigerated spaces by the use of the humidity chart. It is not necessary to devote any great attention to the theory of the chart itself, which has been extensively developed. For our immediate purposes a simplified form will do, as in Fig. 1. It is necessary only to draw in three sets of lines. First, we have the horizontal lines, along which w, the weight of water vapor, is constant, measured as mentioned on the right hand vertical scale in grains per pound of

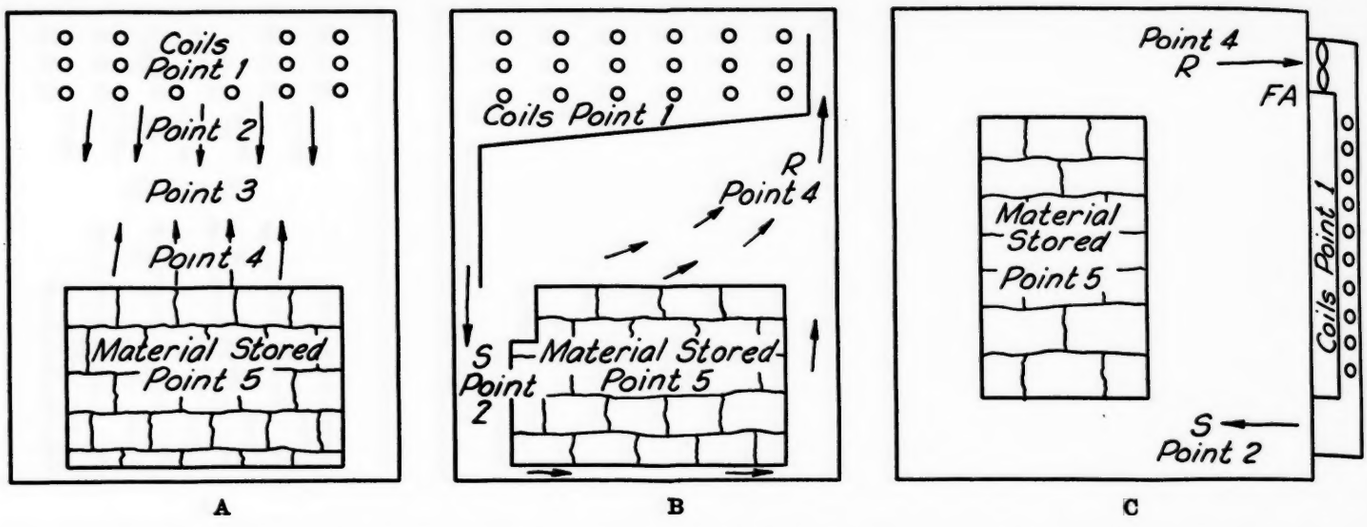

Fig. 2. Air flow in refrigerated spaces. A, conventional gravity flow; B, uniflow, gravity; C, uniflow, forced convection.

air. The chart always refers to one pound of air plus moisture contained. Secondly, we have the family of vertical lines, representing constant temperatures, with a scale at the bottom in degrees F.

As is well known, air will hold a limited amount of water; when it is saturated we speak of 100% relative humidity. Moreover, the ability of air to hold water declines with a fall in temperature. Thus, we must have a line sloping to the left to represent saturation. Such a line as *SS'* of Fig. 1, indicates the maximum moisture held by air at several temperatures. The third set of lines is a group curved like the saturation line, dividing each vertical into ten or more equal parts. This is largely a matter of geometry. In Fig. 1 a line tagged 50% relative humidity divides each vertical into two equal parts. Lines of constant wet-bulb temperature and specific volume are generally drawn on these charts but have been omitted for purposes of this discussion.

Air Cycles in Refrigerated Spaces

The next question is how to plot the cycle of the air in a refrigerated space by means of a chart like Fig. 1. First, let us see how an actual system looks. Fig. 2 shows cross sections of three rooms, (1) in which the air is supplied from outside coils by a fan and (2) where the coils are in the cold space. In all cases the following points apply

Point No.	Designation
1	Air at surface of the coil
2	Supply air
3	Air in cold space
4	Return air
5	Air at surface of goods

The procedure is quite the same, if instead of a coil, we use a spray of cold water, chunks of ice, or any other cooling medium.

It is evident that the coldest point is 1, and the hottest is 5, the others lying between the extremes. Point 3 is best defined as the average of the supply and return air, since in the conventional system of Fig. 2A, it is arrived at by the mixing of cold supply air from the coil, and the rising warm return air from the goods.

Now let us plot these five points on the humidity chart. The result, Fig. 3, is very simple indeed. They all lie on the same straight line from 1 to 5, with point 1 on the saturation line. The following Explanation may serve the purposes at hand: At point 5 we have the conditions at or near the goods stored or refrigerated, which are assumed to provide the chief source of heat and humidity, having the highest temperature in the system. If they have a moist surface, it will be the source of some or all of the atmospheric humidity in the room, so that 5 is at a higher humidity value on the vertical scale than any of the other points. At the other end, point 1 is necessarily of the lowest temperature and humidity of the system. It is of the lowest temperature, because it is the coil surface which refrigerates the air of the room. It is of the lowest humidity because moisture condenses on the coldest surface in any space; warmer air approaching this surface is losing moisture and air which has passed this surface holds the maximum possible amount of moisture at its temperature.

The air is at all times in movement. At point 5 we speak of a temperature-humidity condition referring not to the state of a given mass of air but of whatever particular bit of air happens to be there at the moment. This permits us to think of changes in temperature or humidity simply as a process of mixing two kinds of atmosphere. About the goods we have air described by point 5. To this we introduce air from a supply line of the state described by point 2. This is how we arrive at point 4 in Fig. 2B or 2C. Point 4 simply has the mean humidity and temperature of points 2 and 5, and this means it must lie on a straight line between them. By similar reasoning, air at point 4 gets together with air at point 1 to produce air at point 2.

The Dew Point Rule

This straight line which composes the refrigeration (low side) cycle, is constant as to the ratio of change in humidity, $\triangle w$, to change of temperature, $\triangle t$. That is, the slope of this line measures the relation of humidity lost to temperature lost, as air passes from state 5 to state 1. This defines a certain kind of re-

frigeration situation, always the same under similar circumstances of operation. For example, if one were rapidly pre-cooling a substance containing moisture, so much lowering of temperature, $\triangle t$, would bring about a lowering of water content, $\triangle w$, for each pound of air circulated. On the other hand, were one cooling some other product at a different rate of heat removal, he would get a line of some other slope.

It is frequently the objective to reduce the (surface) moisture loss to the minimum, while maintaining as high a rate of temperature reduction as possible. The rule is "don't let the lowest temperature get below the dew point". This refers to the dew point of the air next to the stored goods, point 5, and this point is at the union of a horizontal with the saturation line, namely point 1′ of Fig. 3.

Operation with a refrigerated surface at a temperature above this point 1′ will take heat out of the air, reducing its temperature, $\triangle t'$, without the removal of any moisture. In other words, there is a state for any atmospheric condition such as point 5, below which you cannot cool this atmosphere without causing precipitation (dew) on the cooling surface, point 1′. If one stays above this dew point there can be no loss of moisture.

If all refrigerator spaces were so designed that the air never got below the dew point, there would be no water or frost accumulation on the coils, and little loss of moisture from the goods. However, this would permit only a very small difference in temperature between the goods and the cold surface, about 4 to 5 F in the usual case. As a result, it would be necessary to compensate by using a much larger surface area than is usually the custom.

In summary then, the humidity chart permits a graphical way of looking at the conditions in a refrigerated space. It is composed essentially of a grid where one direction represents temperature and the other direction represents atmospheric humidity or moisture. On this grid the series of conditions of air as it flows through the refrigerated space may be plotted. This movement is such that air passes a coil or other cold surface at the lowest temperature and humidity in the system, point 1, from which it moves to the surface of refrigerated or stored goods as foodstuffs, at the surface of which are found the highest temperature and humidity in the system, point 5. When points 1 and 5 are located on the chart, a straight line may be drawn between them, and all the other conditions experienced by the air in any part of the system will lie on this line.

Example: In a freezing tunnel where air is circulated over a coil at OF, the temperature in the vicinity of the material being frozen is observed to be 25F and the relative humidity 75%. How much water will freeze on the coil per day, if 100 cfm of air reaches the temperature of the coil? Answer: Using Fig. 1 as a rough basis of computation, we can read off the values of humidity w, at points 1 and 5 as follows:

Point	Temperature	Humidity
1	0	5
5	25	15

This tells us that each pound of air will drop 15—5 or 10 grains of water at the coil as it flows over it. The number of pounds of air flowing per day will be the total volume or 100 x 60 x 24, divided by the volume of air in cubic feet per pound, about 12.5, all of which equals 11,500 pounds per day. This means 115,000 grains or 16.4 pounds of water.

A more complete chart and further examples will follow in other articles of this series.

Let us summarize the major conclusions as follows:

1. For the elementary case considered, points representing the successive temperature and humidity states of the air in circulation in a refrigerated space lie on a straight line, called the cycle line, on the psychrometric chart.

2. Humidity loss always takes place in a refrigerated space, unless the cycle line is horizontal, that is unless the lowest temperature present is above the dew point corresponding to the highest temperature present.

Part 2 — Condition of Supply Air

The term supply air is seldom used in refrigeration, but it fits here just as well as it does in air conditioning to describe the air leaving a coil or entering a refrigerated space. The question at hand is how to figure out the temperature and humidity of this air, knowing the conditions of the air as it enters the coil, and the size of the coil. An accurate answer to this question, as simple as it seems, is by no means easy to find in present refrigeration literature. In fact, in many an existing system, this information was unknown to the man who designed it in the first place.

This article will attempt to find rules which apply, based in the main on the psychrometric or humidity chart, a sample of which composes the upper part of Fig. 4. For this purpose we may take a numerical example, and from the conclusions try to see what would happen if other specific conditions were considered. For example, we can then ask such questions as this: What will happen to the temperature and humidity in a room if we increase the amount of coil surface?

Let us assume the following conditions:

Coil surface area, $A = 120$ sq ft

Coil face area, $A_1 = 3$ sq ft

Coil surface temperature, $t_1 = 33$F

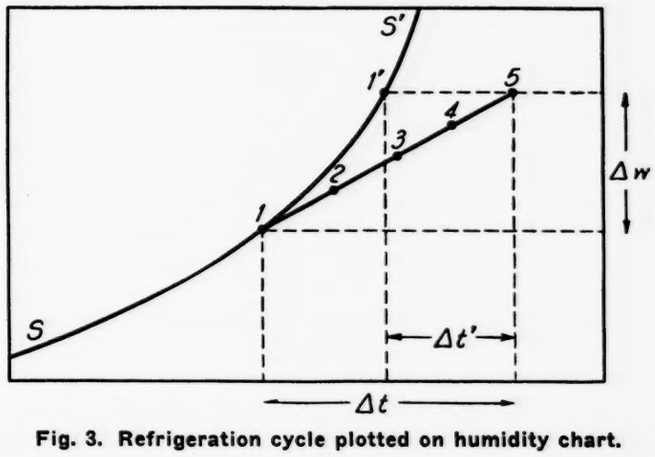

Fig. 3. Refrigeration cycle plotted on humidity chart.

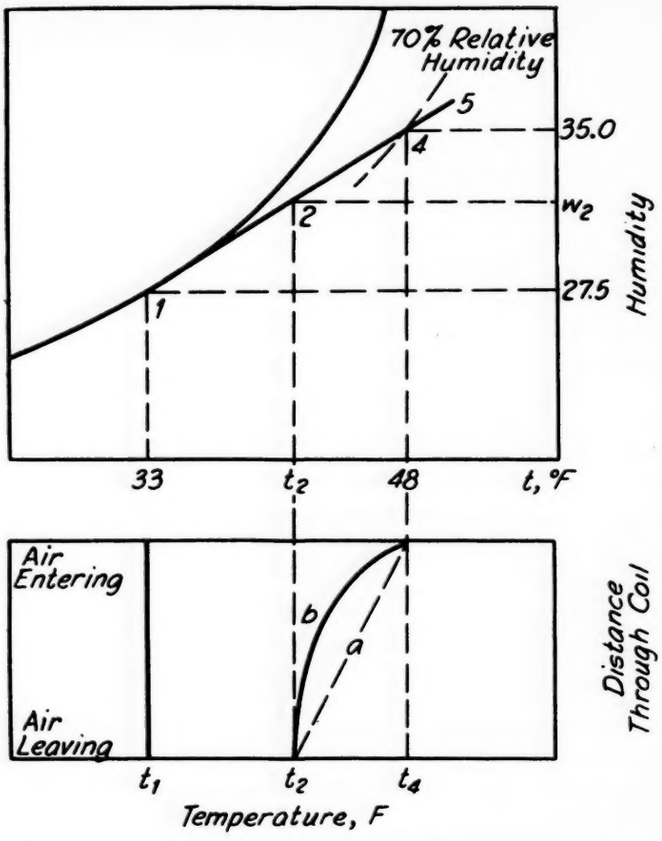

Fig. 4. Cycle line and coil air flow temperature diagrams.

Air temperature entering coil, $t_4 = 48$F
Air velocity, $V = 1200$ fpm
Air humidity entering coil, 75% relative

At the outset we know that point 2 must lie on a straight line defined by points 2 and 4 on the humidity chart, as part of the theory established in Part I that all the points representing a refrigeration cycle lie on a line from 1 to 5. Of course, there are exceptions, as holds with any theory, but this describes the simple essential truth of the situation.

To make the state of affairs inside the cooling coil more graphic, the humidity chart has been extended at the bottom of Fig. 4 by another diagram. The horizontal scale here is again temperature, on which points 1 to 5 may appear. The vertical scale is the distance through the coil, from entering to leaving. It will be noted that the entering air has temperature t_4 and that the leaving air has a temperature t_2, while the coil temperature is represented by the vertical line at t_1.

This diagram goes with the cycle of Fig. 2c. It is more convenient to take an example where the air is under forced convection, that is fan-driven, since the various points are isolated. However, the same principles apply to the case of natural convection. Where we have the cycle of Fig. 2a, we use t_3 instead of t_4 as the temperature of the return air.

Supply Air Equations

In the lower portion of Fig. 4, the curve representing the change of temperature in the coil, from t_4 to t_2, is not a straight line *a* but a curve *b*. Some readers may not know why a curve is to be expected in heat transfer problems of this sort. The reason is that when a fluid transfers heat to a solid surface, more happens at the end where the contact first takes place. It may be visualized thus: If the entering air is at 48F, and the leaving air is at 40F, then the air at the middle of the coil will not be at 44, but at some lower value. More heat will be removed in the first half of the process, merely because the difference in temperature between that air and the surface is higher than it is in the second half. In the general case, simply as a matter of geometry, the average or mean temperature difference is expressed as

$$\text{MTD} = (t_4 - t_2)/[2.3 \log (t_4 - t_1)/(t_2 - t_1)] \quad (1)$$

With this graphical base at hand, let us scramble the various factors into an equation which may be solved for the unknown temperature of point 2. It appears that we have two expressions for the total amount of heat flowing.

In the first place, we may set forth an expression for the heat removed from the air stream, as follows:

$$60\ c\ V\ A_1\ \rho\ (t_4 - t_2)\ \text{Btu per hr} \quad (2)$$

This simply says that each pound of air passing through the coil gives up heat per degree of temperature change, as measured by the specific heat. The specific heat, $c = 0.24$, is virtually constant, in Btu per lb per F. The number of degrees F is t_4—t_2. The number of pounds flowing per hour, for each square foot of face area figures out as 60 V ρ, where ρ is the density of air in pounds per cubic foot, or about 0.08. This boils down to 60 x 0.24 x V x 3 x 0.08 (t_4—t_2), or

$$3.45\ V\ (t_4 - t_2)\ \text{Btu per hr} \quad (3)$$

In the second place, we may set up another expression:

Amount of heat flowing into the coil,

$$= h\ A\ (\text{MTD})\ \text{Btu per hr} \quad (4)$$

where h is the coefficient of surface heat transfer in Btu per (sq ft) (hr) (F) and MTD is as given in equation (1). The value of h is not constant, varying chiefly with the velocity of the air. In general, heat transfer increases roughly as the square root of increases in velocity. Let us for purposes of calculation here make

$$h = \sqrt{V/10} = 0.32V^{0.5}$$

Then we may write equation (4) as:

$$0.32\ V^{0.5}\ 120\ (t_4 - t_2)/[2.3 \log (t_4 - t_1)/(t_2 - t_1)] \quad (5)$$

And then equate the two expressions for the same amount of heat, that is (3) and (5), thus:

$$3.45\ V\ (t_4 - t_2) = \frac{0.32\ V^{0.5}\ 120\ (t_4 - t_2)}{2.3 \log [(t_4 - t_1)/(t_2 - t_1)]} \quad (6)$$

Substituting the value of V = 1200 and transposing, we get:

$$\log [(t_4 - t_1)/(t_2 - t_1)] = 0.1390 \quad (7)$$

Since $t_1 = 33$, $t_4 = 48$, and the number whose logarithm is 0.1390 is 1.37, we find that $t_2 = 43.9$F.

Some General Rules

We have now solved a typical problem relating to the air leaving a coil. Can we now deduce some

broader conclusions? First, let us see what would happen if we increase the air velocity? Will the air leaving at point 2 be colder or warmer?

To find out, we may simply substitute a value of, say V = 2000 in equation 6 and solve. The answer turns out that t_2 now is 45.8. This means that the higher the velocity, the higher the final air temperature. It might be expected that such would be so, because for each increase in flow, more air is in use but the amount of cooling surface is the same as before, while the rate of heat transfer is increased only as the square root of the increase in the velocity.

It must be remembered that if you change one variable you assume the others stay the same. In this example, the surface temperature stays at 33F. In a real case, to increase the air flow is also to increase the load, or the amount of heat, and the refrigerating machine may not be able to carry it off, in which case the temperature of the coil will rise.

By similar analysis, we find that an increase in area or rate of heat transfer will cause a decrease in t_2.

In summary, the supply air temperature in any system will be lower, other things in each case remaining the same, as

a. The coil surface area is increased.
b. The surface rate of heat transfer is increased.
c. The air velocity is decreased.
d. The coil surface temperature is decreased.

Now what about the humidity at point 2? Since we know that the temperature is 43.9, we can locate the humidity along the cycle line of Fig. 4, between point 1 and point 4. It is simply a matter of direct proportion:

$$\frac{35.0 - 27.5}{48.0 - 33.0} = \frac{35.0 - w_2}{48.0 - 43.9}$$

From which we find at $w_2 = 32.9$. It may also be seen from Fig. 4 that while the absolute humidity w is lower, along with the temperature, at point 2, than at point 4, the relative humidity will be higher. This will be more clear if plotted on the full psychrometric chart to be presented in the next article of this series.

In the previous part, we considered the fact that water will be deposited out of the air on the coil (or ice, if the coil temperature is below 32F). There appeared to be one situation where no moisture would be so lost from the air, that is when the lowest temperature is above the dew point. Fig. 5 will make this clearer. The cycle line which will give the minimum moisture loss will be the horizontal line marked Minimum. In refrigeration it is often desirable to avoid loss of moisture, in order to avoid drying the goods stored. On the other hand, in air conditioning it is more often the objective to get rid of the moisture. In this case it is desirable to operate with the steepest line, and such a line, giving the largest yield of moisture on the coil, is shown by the line marked Maximum in Fig. 5, namely the line which is tangent to the saturation curve.

Using the concept of Fig. 5 and the conclusions previously arrived at in the matter of supply air conditions, what may we say of the control of humidity? If we were to repeat the above conclusions, it would seem that like temperature, humidity would become

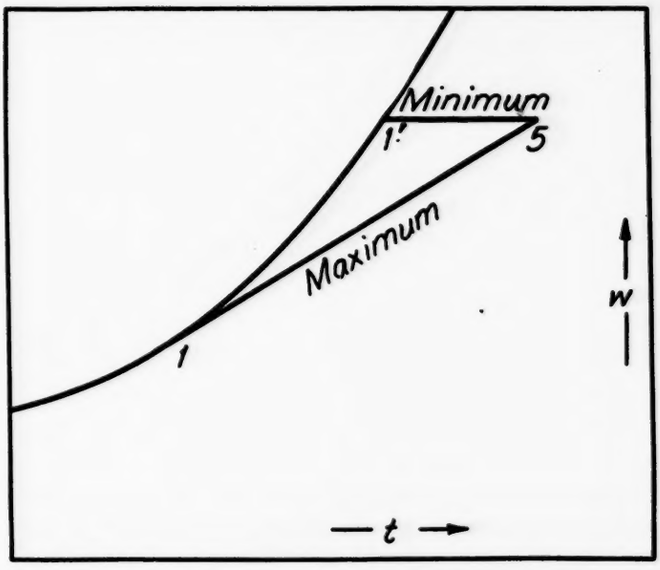

Fig. 5. Refrigeration cycles for minimum and maximum loss of moisture.

lower as we increase the coil area or heat transfer rate, or as we decrease the velocity or surface temperature. But one familiar with refrigeration will at once object to such a conclusion. Is it not true that increasing the coil area, increases rather than decreases the humidity? The statement seems to contradict the usual rule that too small a coil surface is the cause of drying of the goods.

The explanation lies again in the question of what variables are held constant. As noted before, these changes result only if other things remain as they were before. We have really asked this question: What will happen if we add more surface without changing the temperature of this surface? Certainly the result will be the removal of more heat and more moisture too. This lowers both temperature and humidity. The other case, the usual case one thinks of in connection with an increase in surface, is where there is no increase in the amount of heat, or the so-called load. In this case if you add surface you require a smaller temperature difference, with the result that t_1 will come nearer to t_4 and the line from 1 to 4 in Fig. 4 will be more nearly flat, causing less moisture removal, by the reasoning used in Fig. 5. Conversely, when too small a coil is used, the temperature t_1 necessary to meet the load conditions becomes lower, and the moisture removal conditions approach the maximum.

In practice, none of the variables mentioned can be put to immediate convenient use as a measure of control over humidity. The humidity in most refrigerated spaces is a matter of accident rather than intent. The only assurance against conditions that are too dry is to use ample surface, in order to approach the condition where the surface temperature is not too far below the dew point. However, this is more of an ideal than a reality, unless the exact use of the space is foreseen.

In the operation of a given system, there is no common means for varying the area of the coil, the rate of heat transfer or the air velocity, to effect changes in humidity. The coil temperature will vary with the load and the capacity of the condensing unit, and any change which tends to take more heat tends likewise to take more moisture. A scheme of control

proposed by Boehmer reduces the cross sectional area of air flow by variable louvers. This will have the effect of using less of the area of the coil and raising the supply air temperature and humidity (with a reduction also of load).

Conclusions

At the end of part I we formulated two principal conclusions. We may now add two further general rules regarding the behavior of air in a refrigerated space as follows, applicable to a given coil receiving air of a given condition:

3. For a fixed coil temperature t_1, the temperature and humidity of the leaving or supply air t_2, w_2 will decrease with an increase in surface area A or rate of surface heat transfer h, or with a decrease in velocity V.

4. Factors A, h, V and t_1 are not commonly subject to control in order to control humidity, except as they are provided for in the initial design. In an existing system, excessive moisture loss may be reduced only by increasing the coil temperature, but this will be paid for in a reduction in the load or amount of refrigeration accomplished.

Improved Ice Calorimeter in High-Temperature Research

Research on high-temperature specific heats in a National Bureau of Standards laboratory has resulted in the development of a modified Bunsen ice calorimeter comparable in precision of results with the best of the present-day electric calorimeters, and at the same time offering certain advantages over such instruments. The Bunsen ice calorimeter, designed in 1870, furnished at that time a rather crude method for the determination of the amount of heat energy given up by a body in cooling to the temperature of melting ice. Until its application to this project, it had not previously been used for extremely accurate work, largely because it had never been investigated from the standpoint of high accuracy with modern laboratory technique. The instrument, as modified by D. C. Ginnings and R. J. Corruccini, was constructed specifically for use in conjunction with a specially designed high-temperature electric furnace to obtain specific heats and related thermodynamic data at elevated temperatures. In addition, this new ice calorimeter has provided the most accurate means for measuring the density of ice at its melting temperatures, as it is free from many of the difficulties inherent in other methods.

In the application of the ice calorimeter to specific heat determinations, a sealed capsule containing the sample material is first heated to a known temperature in an electric furnace. The capsule is then dropped into the ice calorimeter, where the heat evolved in cooling to the temperature of melting ice is measured.

The ice calorimeter, by measuring volume changes taking place during the melting of ice, measures heat input by weighing mercury. In other words, the heat given off by the capsule and contents is allowed to melt ice that is in equilibrium with water in a closed system, and the resulting volume decrease is determined by means of mercury drawn into the system. The calibration factor is defined as the ratio of the amount of heat energy input to the mass of mercury intake and may be expressed in international joules per gram of mercury. Once the calibration factor of the ice calorimeter is accurately determined, temperature or electrical measuring devices are not required by other experimenters using ice calorimeters; the amount of heat energy added to the calorimeter may always be computed directly from the mass of mercury drawn in and the calibration factor.

The furnace used with the ice calorimeter at the Bureau consists of a vertical alundum tube, 1 inch in diameter, having a chromel winding, and surrounded by insulation. The upper end is stoppered by a fired talc plug, and the temperature within is measured with a calibrated platinum versus platinum-rhodium thermocouple. Proper apportioning of power to auxiliary heating elements, covering a 3-inch length at each end of the furnace core, results in a more nearly isothermal central region, in which the test sample is initially suspended. Tests made at temperatures up to 1,000° C with thermocouple junctions located at various points within the furnace indicated that under optimum conditions gradients of only a few tenths of a degree per inch still existed in the central region in which the capsule was to be suspended.

The calorimeter is provided with a thin-walled central well of low thermal conductivity, into which the capsule containing the test sample is dropped after heating to the desired temperature in the furnace above. The lower part of the well is surrounded by two co-axial glass vessels, which provide an insulating space between an inner ice-water system and a surrounding ice bath. The space between the central well and the inner glass is filled with air-free water, which is partially frozen at the beginning of an experiment. Any change in volume due to melting ice in the inner vessel is transmitted to a mercury reservoir, resulting in a change in the amount of mercury in a weighed beaker. An electric heater sheathed in a metal tube is soldered on the outside of the central well to introduce heat for calibration purposes.

Radiation of heat from the interior of the furnace downward into the calorimeter is prevented by a specially designed gate, which may be opened for a short time to prevent dropping the capsule. An arrangement is also provided for the constant flow of dry carbon dioxide, previously cooled to the temperature of melting ice, upward through the central well to prevent diffusion of water vapor from above.

A series of electric calibration experiments was carried out to determine the effect of a number of variables, such as the freezing rate and aging of the ice mantle formed about the central well, heat input, carbon dioxide flow rate in the central well, and pressure in the closed system. It was found that these variables had little if any significant effect on the calibration factor.

Possibilities of Energy Storage in Heat Pump Applications

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This is a discussion of how energy storage has been utilized in some applications of the heat pump to meet short periods of maximum heating demand.

THE sizing of any type of home heating equipment is based on the maximum demand. This maximum demand is encountered only a few days a year and in many cases only for a few hours in each year. Applications of the heat pump, particularly in areas where the heating requirements exceed the cooling requirements, must be based on meeting the requirements of these few days. If, through the medium of a storage system or through other means, this peak can be cared for, many benefits would result. The home owner could use a smaller, hence a lower cost installation and the power company would benefit through lower peak requirements and a higher load factor. These latter items are likewise important to the home owner since they would undoubtedly result in more favorable power rates.

Before going into factors which influence the utilization of storage in connection with the heat pump, it might be well to consider the work which has been done in connection with heat storage in the past, both to see the type of equipment which has been used and its limitations, and how this experience can be used to advantage in developing new types of heat storage equipment. The possibility of storage of energy for heating has for many years received the attention of electric utility engineers. Until the successful development of the heat pump, energy storage represented the only hope of the electrical industry to participate directly in the home heating field.

Most of the early electricity for heating developments attempted to make direct electric heating competitive with conventional furnace systems by the development of off-peak heat storage systems. These developments were largely unsuccessful due to (1) the bulky equipment necessary for storing the large quantity of heat required, and (2) the demand (over twice the normal residential demand) which would necessitate large distribution equipment, and (3) the high cost of energy.

There are a number of installations reported in technical literature which use direct electric heating to charge a storage tank. The heat stored during the off-peak periods is then used for heating during the following day. All installations listed use either water or an oven filled with rock or other inexpensive material as a heat storage medium. Three of these installations, two using water and the other granite boulders, are described. These installations while not generally satisfactory, do give some indication as to the size and operation of the heat storage tank.

Storage Systems Using Direct Electric Heating

A rock storage oven using off-peak power for building heating was constructed at Mason City, the con-

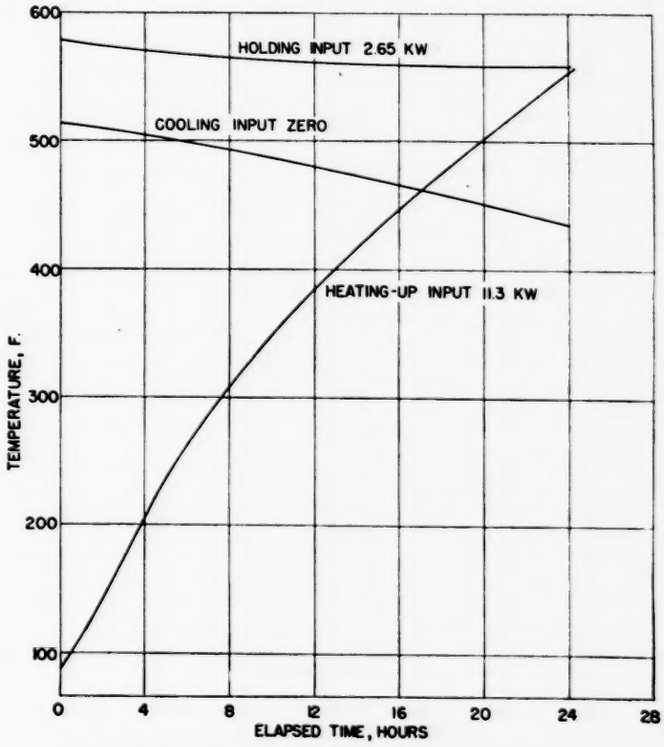


Fig. 1. Results of tests made on a rock oven at Grand Coulee Dam.

Part of the material for this article was taken from a report prepared for Southeastern Electric Exchange in connection with their project at Southern Research Institute.

tractor's camp at Grand Coulee Dam, on which tests were conducted by the State College of Washington. Granite boulders from 3 to 60 lb were used as a storage medium. The oven was constructed with a 4 in. interior wall of brick with a tile wall 3 in. behind it. The space between the brick and tile walls was filled with rock wool. Two inches of diatomacious earth were placed on the floor to insulate the oven from the ground. The oven was designed for storing 1,000,000 Btu at 600F with a minimum temperature of 170F. Fig. 1 shows results of tests made on the rock oven.

An experiment was conducted in a residence at LaGrange, Ill., by Public Service Co. of Northern Illinois in 1928, using water as a storage medium for off-peak energy. The storage system consisted of five 200-gal water tanks, four of which were ample capacity to provide heat for the dwelling. Each tank has two 5-kw 220-volt immersion heaters. Water tanks are located in an insulated heat room and the building is heated by blowing air through the room over the water tanks. The water was heated to 282F on some days. However, it was found that 265F was sufficient to provide 30% reserve for the most severe conditions. The coldest temperature encountered during the test was —16F. Fig. 2 shows data on this water storage system.

Off-Peak Heating

There have been several other systems, the most notable being the installation made by the Hall Electric Heating Co. at Ft. Wayne, Ind. These systems consisted of a number of tanks filled with fluid which was heated during off-peak periods and were then used to supply heat to the building through a conventional hot water or forced hot air distribution system. The fluid was heated to approximately 300F by direct electric heaters. Some results of the investigations made at Ft. Wayne are given in Table 1.

An installation shown schematically in Fig. 3 has been in operation in the Magella Sub-Station of Alabama Power Company, Magella, Ala., since 1924. During the period of power shortage this installation operated on off-peak power, being charged from 9 p.m. to 6 a.m. Eight 5 kw immersion heaters were used in the tank. A ¼ hp circulating pump was used to distribute the hot water to the heating system as required. This installation utilized two tanks 5 ft in diameter by 7 ft long which were charged with water at 10 lb pressure to 240F. The installation heated the 70,000 cu ft Sub-Station building. This installation by Alabama Power Co. was apparently the first in this country and is still operating successfully. It was designed by W. R. Waggoner and W. E. Mitchell, both then of the Alabama Power Co.

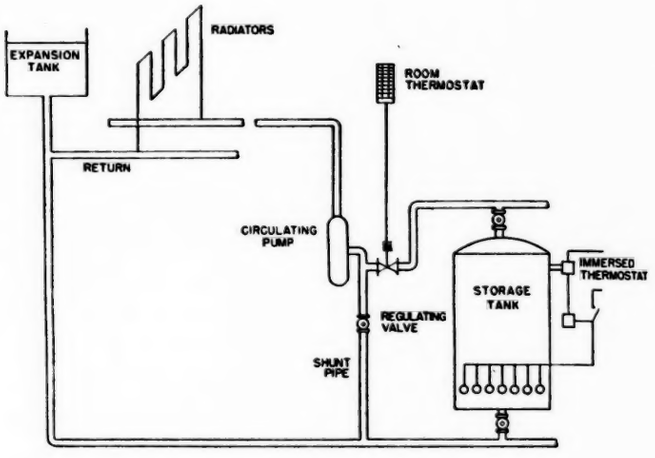


Fig. 3. Storage tank installation at Magella, Ala.

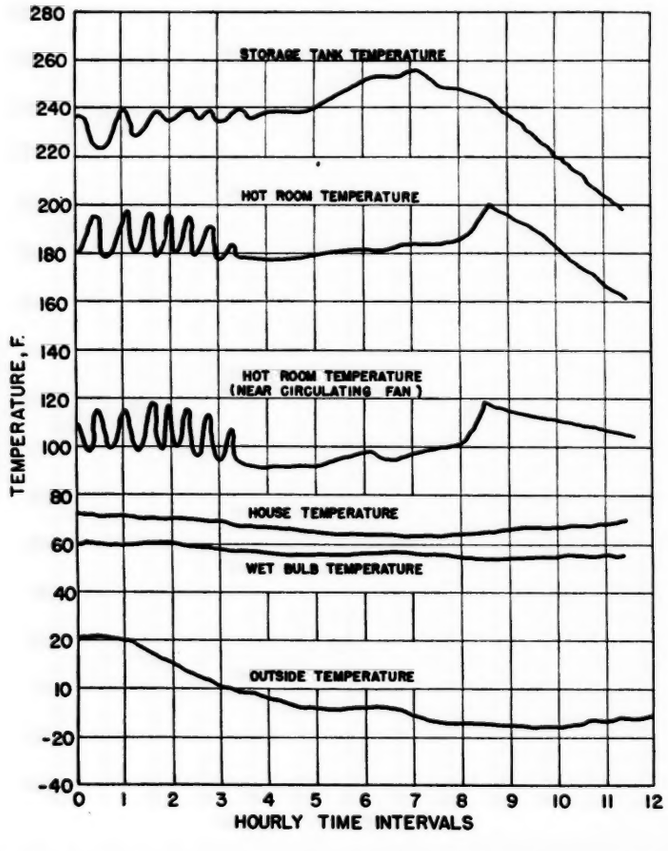


Fig. 2. Data on a water storage system in northern Illinois.

Storage Tanks

One of the problems in energy storage is the large size and relatively high initial cost of the heat storage tank. For example, a house having a heat loss of 45,000 Btu per hr to be heated with a storage tank would require a tank capacity of 7500 lb of water if the tank is to be charged in 9 hr and the heating is be spread over the equivalent of 15 hr. Using resistance heaters, higher temperatures and pressures could be used and advantage could be taken of the latent heat of vaporization to reduce the size of the tank. For use with a heat pump, the water could not be heated much above 120F without seriously reducing the performance of the heat pump. Several substances have been studied to see if they would reduce the heat storage tank size. However, the nature of the dependent variable involved is such that an increase in

Table 1 — Energy Used for Electrically Heating 18 Houses, October 15 — May 15

House	Description of House	New or Old House	Insulation in Operation	Cubic Content Heated	Kilowatts Installed	Net Kilowatt Hours Used*
1	Brick veneer	Old	Walls and ceiling	44,840	80	122,310
2	Brick veneer	Old	Ceiling and double glass	32,499	60	85,970
3	Brick veneer	New	Walls, ceiling and double glass	28,255	50	35,695
4	Clapped frame	Old	Walls, ceiling and double glass	28,124	50	78,750
5	Stucco frame	Old	Walls, ceiling and double glass	26,250	60	70,615
6	Brick veneer	New	Walls and ceiling	22,650	40	35,833
7	Brick veneer	New	Walls and ceiling	22,400	60	51,443
8	Brick veneer	New	Walls and ceiling	21,920	60	55,146
9	Stone	Old	Ceiling and double glass	21,277	60	68,817
10	Clapped frame	Old	None	20,020	50	60,690
11	Clapped frame	Old	None	20,020	50	54,922
12	Clapped frame	Old	Walls and ceiling	15,700	40	37,800
13	Clapped frame	Old	Walls and ceiling	15,620	50	40,025
14	Brick veneer	New	Walls, ceiling and double glass	14,000	30	25,222
15	Shingle frame	New	Walls, ceiling and double glass	13,690	30	13,900
16	Brick veneer	New	Walls, ceiling and double glass	13,170	30	19,080
17	Clapped frame	New	Walls, ceiling and double glass	12,420	30	14,990
18	Shingle frame	New	Walls, ceiling and double glass	9,130	30	17,170

*Some variation between size of house and kilowatt-hour consumption may be accounted for by house temperatures maintained by individual householders, and other unusual operation conditions.

one is counterbalanced by a decrease in the other which tends to equalize the effectiveness of various materials considered in their applicability to heat storage.

Lead, for example, has a relatively high specific gravity of 11.3 which would tend to decrease the volume of the storage medium. However, its low specific heat of .03 tends to equalize the volume of lead and water for the same heat storage capacity. Another consideration was to use a substance with a low melting point and to take advantage of its latent heat of fusion. Here again the essential properties influencing heat capacity equalize each other so that the lower the melting point the lower the value of the latent heat of fusion with the energy storage capacity remaining essentially constant. Water still seems to be the most satisfactory substance for storage systems since it has a high specific heat and is inexpensive. For direct heating, where fluid temperature may be raised above the boiling point, the size of the tank may be reduced by taking advantage of its latent heat of vaporization as was indicated previously. Further study of possible fluids for this type of service would appear to be desirable.

The heat pump opens up new possibilities of application of energy storage since heating requirements are so critical and the maximum demand determined by conditions which exist only a few days or hours out of the year. Such storage has possibilities in reducing equipment size. Perhaps of greater importance as was indicated previously, is its possible effect on increasing load factor which, in turn, would increase the attractiveness of this type of load to the utilities.

Heat Pump Used with a Storage System

There are several heat pump installations listed in the literature which utilize heat storage. However, no operating data are available to give an indication

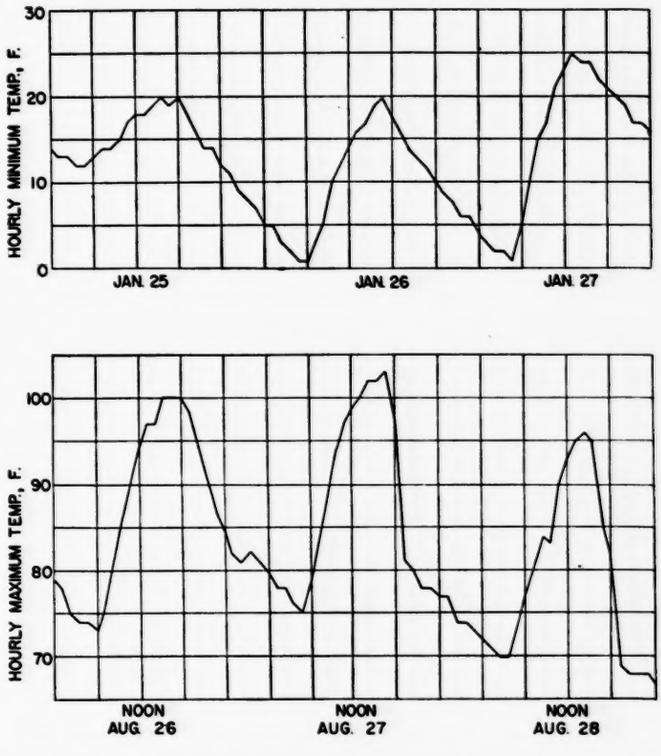


Fig. 4. (Above) Hourly minimum temperature for Birmingham, Ala. Fig. 5. (Below) Hourly maximum temperature for Birmingham, Ala.

of their performance. Some of these storage tanks were connected to provide additional capacity in case the heat pump output was lower than expected and the tanks have not been used. The commercial heat pump installation at Steubenville, Ohio, utilizes a storage tank charged to 110F by one of the compressors in the system when outside temperature is between 40F and 25F. If the temperature falls to 15F water from the tank is used to supplement the output of the heat pump. This system operates wherever the temperature falls to 40F and is not limited to off-peak. The heat pump installation at the United Illuminating Co. has a water storage tank which may be heated during off-peak periods to supply heat the following day. It has never been necessary to use this storage tank since the heat pump installation is oversized for the present heating requirements. The Ohio Power Co. is installing a heat pump with a storage tank in its Brilliant, Ohio, office building. At present no data are available on this installation.

There are several possible methods of adapting the heat pump to a heat storage tank for heating. (1) The heat pump may be used to charge the tank during off-peak periods with the tank supplying all the heat during the load peak hours. (2) The heat pump may operate continuously, charging the tank during the night. The normal heat output may then be supplemented by the storage tank when required. (3) Supplementary electrical heat may be supplied by a storage tank which is charged by direct heaters. (4) A combination of off-run heat pump storage and supplementary heat may be used. (5) Supplemental heat may be used to carry the peak requirements although it is not as desirable because of the poor load factor.

In considering the first of these possibilities it is obvious that a large storage tank would be required if all of the heating requirements were supplied by the storage tank. This is especially true with a heat pump since the water could only be charged to about 120F and still maintain a high COP for the heat pump. In addition, the size of the heat pump would necessarily be increased since, for example, the heat pump would make up for the heat loss for 15 hr while the storage tank charging time would be only 9 hr. Such a system would only be applicable where the off-peak power rates were very low.

In the second case, the size of the storage system as well as the size of the heat pump could be substantially reduced since heating requirements could be supplied over a longer period of time. In addition, cold water for use in summer cooling might also be stored which would be of particular advantage where summer and winter loads are merely balanced or where summer loads exceed the winter heating requirements.

Fig. 4 shows the minimum hourly temperatures for Birmingham, Ala., for the 3-day period of January 25, 26, and 27, 1940. Although temperatures shown are probably lower than the usual design temperatures, the curve shows typical temperature variations. Assume, for example, that the lowest temperatures shown are the design temperature conditions. It may thus be seen that sufficient capacity exists above 10F to supply the minimum temperature conditions, provided the excess capacity is used to charge some storage system. This will allow the heat pump to be sized based on 10F and will permit the use of a smaller machine than if no storage is used.

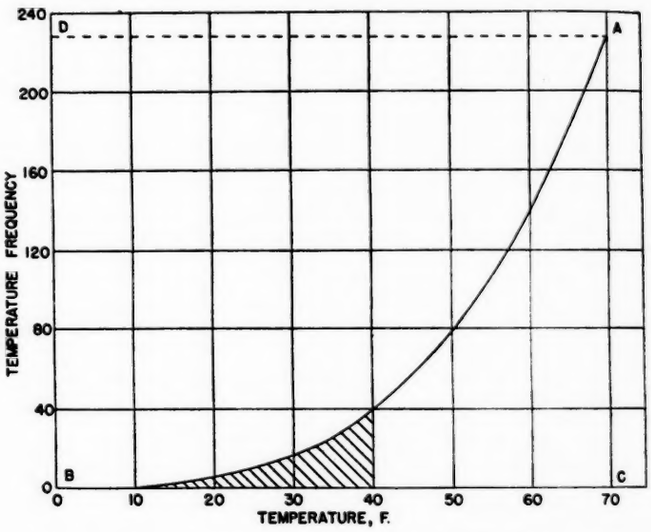


Fig. 7. Temperature frequency curve based on Fig. 6.

In a similar manner, storage of a cold liquid could result in a reduction in the size of the heat pump. Fig. 5 shows the maximum hourly temperature for Birmingham, Ala., for August 26, 27, and 28, 1938. The maximum temperature experienced during this period was 103F. By the utilization of a storage tank, the heat pump could be sized for 90F and still provide ample capacity for the maximum conditions to be experienced.

The third system using a storage tank charged with direct electric heaters, also has possibilities for reducing heat pump size. It substantially decreases the storage tank size by permitting the tank to be charged to a higher temperature than if it were charged by the heat pump output. It has the same advantage as the second system in permitting the use of a smaller heat pump. This is a definite asset to the customer in reducing bulk, hence the initial cost of the heat pump.

A variation of this system is one where the heat pump is used to charge to a temperature of 120 or 140F. Direct heating could be used for the remaining requirements. Construction complications would prob-

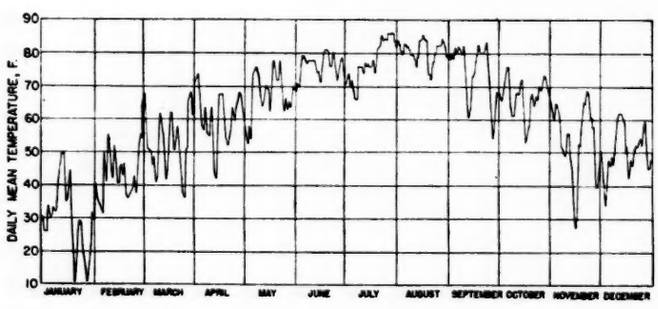


Fig. 6. Daily mean temperature for Birmingham, Ala.

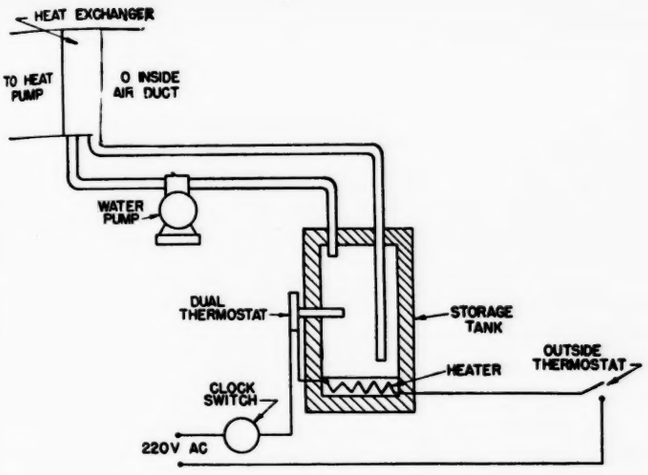


Fig. 8. Arrangement for a two-heating-system combination.

ably eliminate such a combination in favor of possibilities (2) or (3).

The fifth possibility would give the simplest type of auxiliary or supplemental heating systems but it would be more expensive to operate because of the greater energy cost. Standby service, whether electrical or other form of energy, would of necessity be charged for at a higher rate or on the basis of demand. While such a charge would increase the cost of heating, it may in many cases be desired by some customers and should be investigated.

Application Considerations

To analyze advantages of storage systems, it is necessary to determine temperature variations which are likely to occur. Fig. 6 shows the daily mean temperatures for Birmingham, Ala., for 1940. It may be seen that the minimum temperatures shown occur very infrequently. Thus if the heat pump were designed based on this temperature, its total heating capacity would be required only during a relatively short period of time.

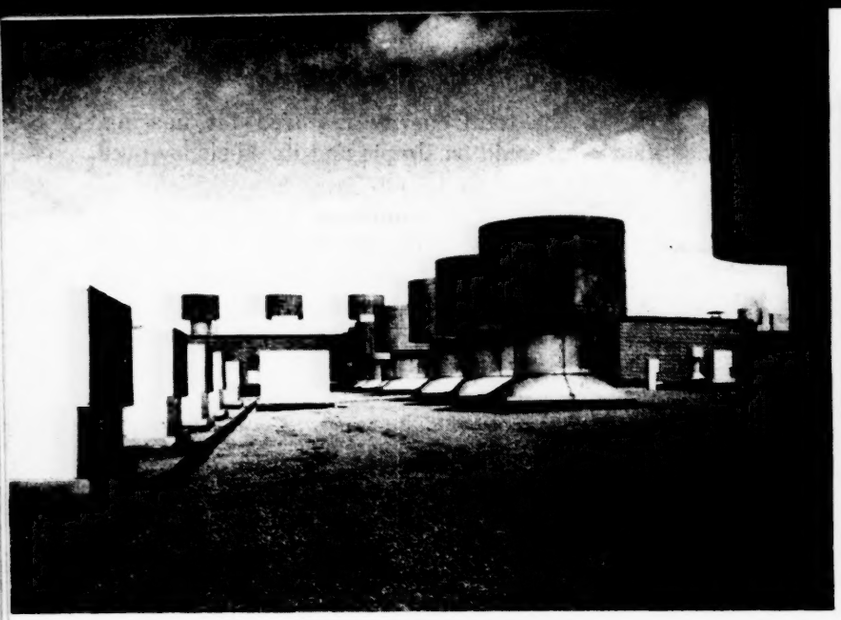
Fig. 7 shows a temperature frequency curve which was constructed based on Fig. 6. This curve consists of a plot of the daily mean temperature against the number of days that the temperature was below that value. Point *A*, for example, indicates that there were 228 days in which the daily mean temperature was below 70F. The area under the curve is proportional to the heating requirements so that the degree of utilization if the equipment design is based on 10F outside temperature and 70F inside temperature, and is given by the ratio of the area under the curve to the area in the rectangle *ABCD*. For Birmingham in 1940 the percentage of utilization for a heating system calculated by this method would be 24%. If the design temperature were 30F., and the additional heat were supplied by a supplementary heating system, the degree of utilization would be 41%.

The area under the shaded portion of the curve represents the total energy which the supplementary heating system would be required to supply. For this area the supplementary heating system would supply only 3.1% of the total energy required for heating. Hence its design could be simple and its efficiency need not be high.

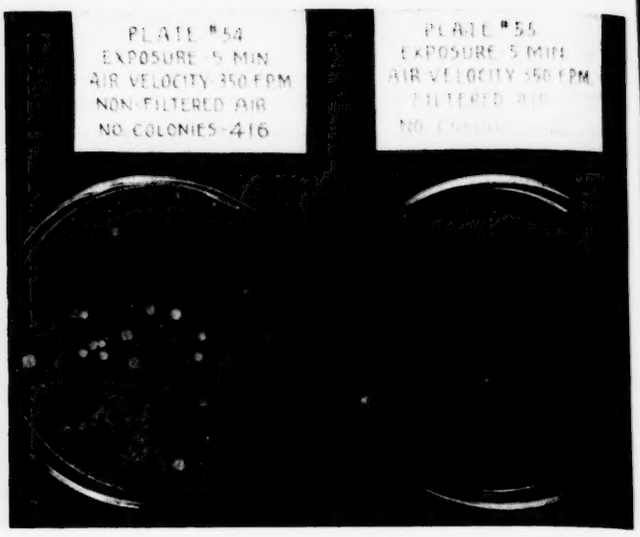
The advantage of the combination of the two heating systems is the increase in the degree of utilization of the larger system, hence a reduction in the initial cost of the equipment. This represents a considerable saving in the case of the heat pump where the first cost is high as compared to other heating systems. The relative size of the two systems depends on the economic balance between the cost of operation and installation of the two systems. Since the maximum demand from the supplementary heating system of this type would be relatively high and its duration and occurence would be short, this heat is supplied from a storage tank charged with off-peak energy.

Fig. 8 shows the arrangement of such an off-peak system. Operation of the electric resistance heaters is controlled by a thermostat located on a storage tank, an electric clock switch and an outside thermostat. Thus when the outside air temperature falls to 20F, for example, and the clock switch is set at a time so that it would operate when there is no peak on the utility system, the electric resistance heaters would charge the storage tank to the value set on the thermostat on the storage tank. During the following day the tank is discharged, supplying heat to the system through a heat exchanger located in the air duct system. A thermostat located on a storage tank prevents the circulation of the water when the storage tank is not charged.

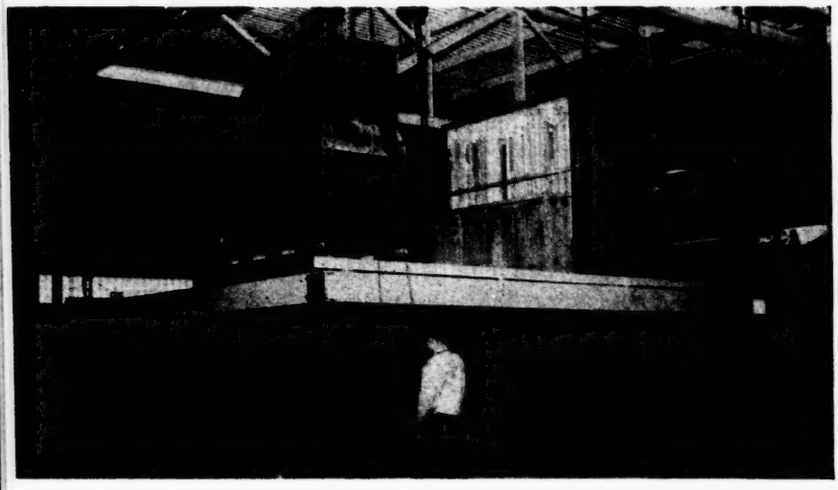
To indicate the cost of operating the storage tank, the following example is shown. A home or building in the vicinity of Birmingham that would require a 3 hp heat pump would have a heat loss approximately 580 Btu per hr per degree difference in temperature. If the design is based on 10F outside temperature and 70F inside temperature, the heat loss is 34,800 Btu per hour. By the use of a supplementary heating system, the design temperature for the heat pump would be raised to 30F so that the heat to be supplied by the heat pump would be 23,200 Btu per hour. The size of the heat pump then could be reduced from a 3 hp to a 2 hp size by the use of these supplementary heaters. This would, of course, result in less expensive equipment to accomplish the heating. By using the degree-day tables for this area, the amount of heat required for a building with a heat loss of 580 Btu per hour per degree was estimated at 37,600,000 Btu per year. With energy cost at 2c per kwh the cost of operation for the supplementary heater is $6.80 per year. The size of the heaters would be 3.4 kw. Since some demand charge would be necessary if these were used in any quantity the cost would be slightly higher. Thus it can be seen for a relatively low operating cost per year the size of the heat pump may be reduced from 3 hp to 2 hp. The advantage of the addition of the supplementary heating system would, of course, lie on the economic considerations between the savings in the size of the heat pump and in the cost of the supplementary heating equipment. Additional advantages are that this would supply a better load factor to the utilities; hence it could conceivably result in a lower energy cost to the customer.



Hundreds of supply and exhaust fans, the largest of which are 100 in. in diameter and 17 ft high, provide ventilation of those departments of General Electric's new Coshocton plastics plant which are not fully air conditioned.



Ability of filters to remove bacteria from forced air in circulation system is demonstrated by tests at Research Products Corp. Plate at left shows results of exposure to non-filtered air; plate at right, filtered air.



Fans and air washers in the air conditioning system at the F. Ducharne Silk Co., N. J., were installed by Austin Co. on a platform at floor level, then hoisted into position. A material reduction in time required for installation is reported for this system.

Picture Paragraphs

A window in a jet engine tube at Shell's Wood River Laboratory permits observation of burning characteristics and patterns of air turbulence during combustion of various types of fuel. Better jet fuels are the object of experiments.

(Left) The 18-story Irwin Keasler Building, Dallas, is being completely air conditioned with 75 five-ton Chrysler Air-temp packaged units.

Steam Vs. Co_2 for Fire Extinguishment in Kitchen Range Hoods

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Chief, Air Conditioning Division, Abbott, Merkt & Co.,
New York

This is a balancing of factors influencing choice between steam and CO_2 extinguishing systems as fire protection for large kitchen range hoods and duct systems. Piping and valve systems meeting requirements of various regulating bodies are described.

RECENT hotel fires with their appalling loss of life and property drive home the vital importance of adequate, dependable and quick acting fire protection. National Fire Protection Association (NFPA) statistics show that 11% of all fires in hotel and restaurants are caused by ignition of grease on kitchen ranges or in the ducts of their hood exhaust systems. Statistics further show that this 11% may be separated into 48% caused by grease on the ranges, 15% for ignition of grease in their exhaust ducts, 9% for special hazards and a balance of 28% due to heat from the range igniting combustible material. Such material is sometimes left temporarily or permanently too close to the fire hazard.

Impressed by such facts, as well as by their own records, the New York City Fire Department took action about the first of this year by issuing the rules as given herein. These rules make steam or CO_2 fire extinguishment systems for range hoods and their exhaust ducts mandatory for hotels within New York City. The fire department is concerned mainly with the prevention of loss of life, hence for the present, rules are confined to hotels only. About the first of next year such rules will be extended to include clubs and possibly restaurants. The subject is a hot proposition in many other places and it appears that similar rules are being or will soon be issued in various cities. In the meantime, the New York City Fire Department has placed violations on many hotel properties due to range hood exhaust systems not in compliance, together with an order to remove the violations within ten days.

To inform the contractor and engineer about these late developments, their detailed requirements (which are none too clear) and the possibilities in this class of business, is the purpose of this article. A further purpose is to show that a fire extinguishing steam system is (1) manual and not automatic, (2) sometimes not possible or practical, (3) sometimes not capable of meeting the fire department requirements, (4) directly opposed to underwriters requirements as to the extinguishing medium used, and (5) does not give the added protection of an underwriters-approved CO_2 system. It will also be shown that an underwriters-approved CO_2 system is not so expensive as commonly thought, in fact, is often as reasonable in cost or may even be less in cost than a steam system for the same job. (See Table—Col. 1-3 for Fig. 1).

Comparative Costs

In actual practice, cost, rather than degree of fire protection afforded, too often determines selection of the particular type of fire extinguishing system. Such basis for selection confines it to the steam system wherever possible, because the owner is more concerned with removing the violation than he is with adequate fire protection. He is, as a rule, unfamiliar with the possibilities in the way of CO_2 systems and is also under the mistaken impression that because steam is already there it can be utilized and extended for the purpose in mind with least expense.

It is, therefore, of more interest to present the matter of cost comparisons first, leaving the relative advantages and disadvantages of steam systems and CO_2 systems with respect to their method of operation for later discussion. Accordingly, the sketches (Fig. 1-2) are presented as a basis for comparison of costs of recent installations in various hotels within New York City.

Surprising enough are the figures in the table—even to the manufacturers of CO_2 systems! This is because the manufacturer usually makes comparisons unfavorable to his own system by assuming that the steam system has a steam supply at the required pressure close to the range, and such is not often the case. The steam system, however, will be the most economical in first cost, only when it combines the most favorable conditions for its installation with a design not comparable to what the CO_2 system gives in the way of added protection.

The fire department does not require remote control on a steam system to stop the exhaust fan at the time of a fire. Where a remote control feature is incorporated, there is an added cost as included in the Table, Col. 2-3-4.

When a fire occurs, the extinguishing medium may extinguish more than desirable, such as the gas flame at the range burners, thus creating a new hazard in the ducts and kitchen. This seems most likely with the steam system dripping condensation and grease from the hood to the range. The fire department, however, has no rule as to automatic closure of the gas supply, but the underwriters do. This, again, increases the cost of the underwriters-approved CO_2 system to the given figures in the Table-Col. 3.

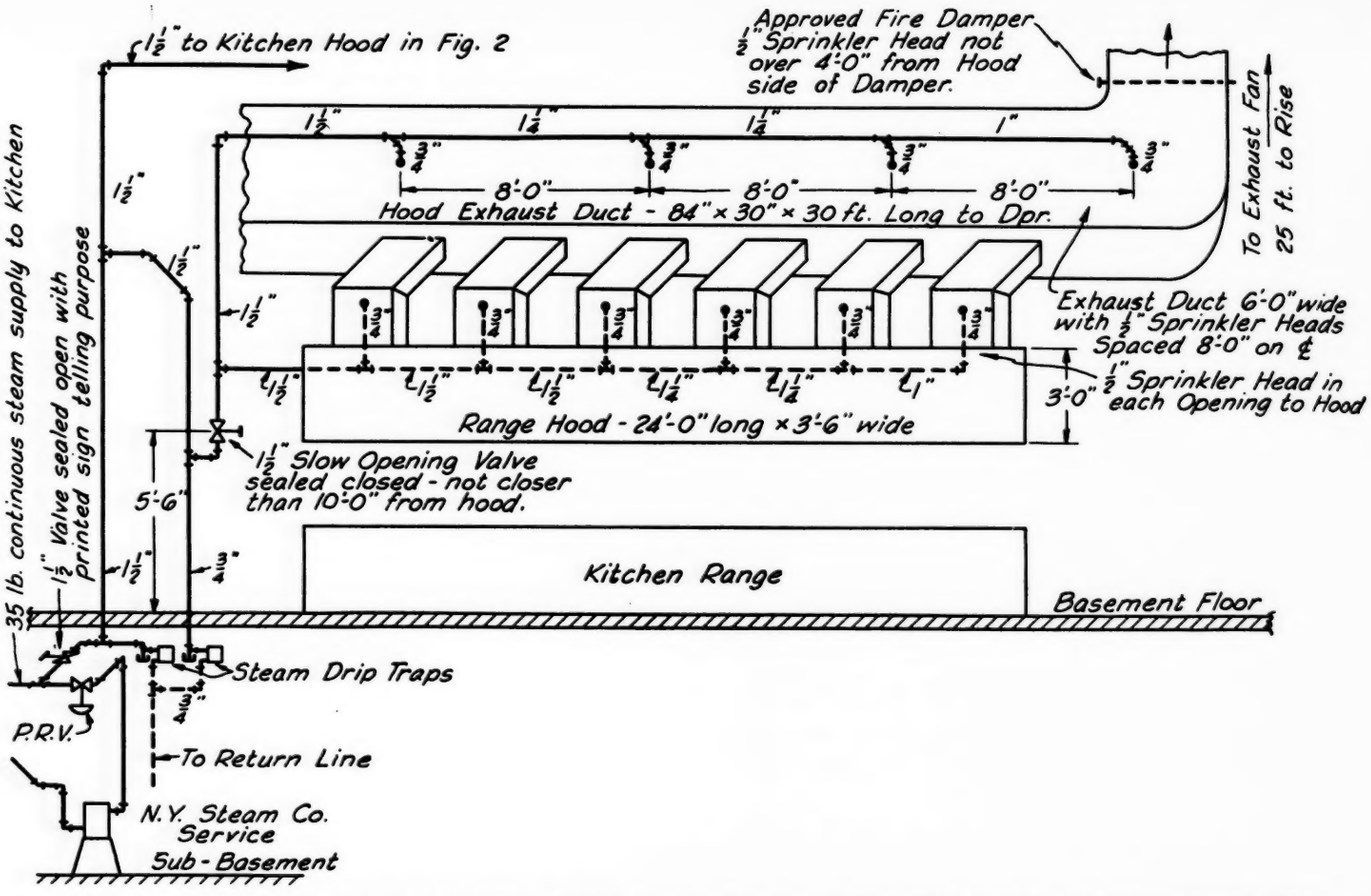

1½" to Kitchen Hood in Fig. 2
Approved Fire Damper
½" Sprinkler Head not over 4'-0" from Hood side of Damper.
To Exhaust Fan 25 ft. to Rise
8'-0"
8'-0"
8'-0"
Hood Exhaust Duct - 84" × 30" × 30 ft. Long to Dpr.
Exhaust Duct 6'-0" wide with ½" Sprinkler Heads Spaced 8'-0" on ℄
½" Sprinkler Head in each Opening to Hood
3'-0"
Range Hood - 24'-0" long × 3'-6" wide
1½" Slow Opening Valve sealed closed - not closer than 10'-0" from hood.
5'-6"
35 lb. continuous steam supply to Kitchen
1½" Valve sealed open with printed sign telling purpose
Kitchen Range
Basement Floor
Steam Drip Traps
P.R.V.
To Return Line
N.Y. Steam Co. Service
Sub-Basement

Fig. 1. Elevation drawing of steam smothering system for a New York City hotel to be in compliance with fire department violation order of June 3, 1947.

Another item of cost in the underwriters-approved CO_2 system is their requirement that the automatic damper must be at the end of the horizontal duct run just before the exhaust riser (See Fig. 3). To protect the usual, long, horizontal, exhaust duct against fire by filling it with CO_2 at the time of a fire costs something for the additional gas cylinders and nozzles. Nevertheless, such protection seems most desirable. For example, a range or hood fire may flash by a damper close to the range before the damper can close, thus igniting the grease laden duct beyond the damper.

If the underwriters-approved CO_2 system is reduced from their requirements for comparison with the economics of a steam system, the cost of the CO_2 system will go down accordingly. In the Table—Fig. 2, Col. 5—the cost of the modified underwriters CO_2 system is about the same as given in Col. 1 for the steam system. Even when this CO_2 system is in this way reduced to the same terms as a steam system, it still gives more protection. For example, a reading of Rule 2 of Underwriters Requirements calls for more than flooding of the duct. It calls for screening of the face area of the hood with gas. The gas within the duct cannot readily disperse, but at the hood bottom it is important that additional nozzles be installed to supply gas for that escaping. This prevents re-flash from a hot range to a grease laden hood.

In all fairness to the costs given for the steam system it should be pointed out that the steam piping from the source to each range in Fig. 1 and 2 was about 150 ft in length. This in itself forms a good portion of its total cost, yet such runs are not uncommon. Actually, the cheapest system of all as shown by the table is the fire department CO_2 system, though it still leaves unsatisfied the underwriters' requirements and, in the writer's opinion, too many hazards that

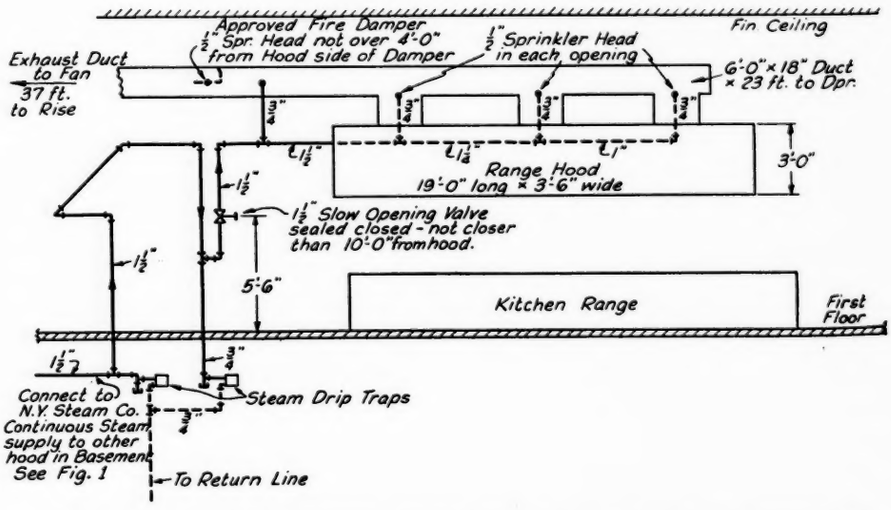

Approved Fire Damper
½" Spr. Head not over 4'-0" from Hood side of Damper
1" Sprinkler Head in each opening
Fin Ceiling
Exhaust Duct to Fan 37 ft. to Rise
6'-0" × 18" Duct × 23 ft. to Dpr.
Range Hood 19'-0" long × 3'-6" wide
3'-0"
1½" Slow Opening Valve sealed closed - not closer than 10'-0" from hood.
5'-6"
Kitchen Range
First Floor
Steam Drip Traps
Connect to N.Y. Steam Co. Continuous Steam supply to other hood in Basement See Fig. 1
To Return Line

Fig. 2. Steam smothering system in a New York City hotel to comply with fire department violation order of June 3, 1947.

Table 1—Comparative Costs of Various Extinguishing Systems

Fig. No.	Medium	Column Number 1	2	3	4	5
		Steam	CO_2			
	Rules	Fire Dept.	Fire Dept.	Approved Underwriters	Modified for Comparison (Not Approved) Underwriters	Underwriters
1	Material	—	$425	$ 663	$ 597	$ 555
	Labor	—	$409	$ 693	$ 610	$ 610
	Total	$1800	$834	$1356	$1207	$1165
	No. Cylinders	—	2	3	3	3
	Cu Ft in Systems	777	777	1215	777	777
	Lbs of Gas	—	98	136	109	109
2	Material	—	$252	$ 491	$ 425	$ 383
	Labor	—	$206	$ 493	$ 410	$ 410
	Total	$800	$458	$ 984	$ 835	$ 793
	No. Cylinders	—	1	2	2	2
	Cu Ft in System	372	372	650	372	372
	Lbs of Gas	—	47	88	71	71

Col. 1—Steam system to meet New York City Fire Dept. requirements includes automatic damper near hood; omits automatic gas shutoff valve and remote control of fan.

Col. 2—CO_2 system to meet New York City Fire Dept. requirements includes automatic damper near hood and remote control of fan; omits automatic gas shut off valve.

Col. 3—CO_2 system to meet National Fire Protection Association requirements includes (as in Col. 2) remote control of fan, but goes beyond NYFD requirements by also including automatic gas shutoff valve and by locating automatic damper far from hood near exhaust riser.

Col. 4—CO_2 System, reduced from NFPA-CO_2 requirements (Col. 3) to give no* greater protection than NYFD-CO_2 system (Col. 2) by moving automatic damper back near hood and omitting automatic gas valve, but including remote control of fan. Such a system would be approved by NYFD, but not by NFPA. Given only for comparison of protection afforded.

Col. 5—CO_2 System, still further reduced from NFPA-CO_2 requirements (Col. 3) to give no* greater protection than NYFD Steam System (Col. 1) by moving automatic damper back near hood and omitting both automatic gas valve and remote control of fan. Such a system would not be approved by either NYFD or NFPA. Given only for comparison of protection afforded.

Note: Labor costs increased 50% to include overtime rates. Kitchen routine cannot be upset in regular hours. There are, however, some installations where work can be done in regular hours, such as in cafeterias. In others, a part of the work, such as the setting up of cylinders in racks need not necessarily be done at overtime rates.

Example: By NFPA rules a combined duct and hood volume of 777 cu requires (777 cu ft ÷ 16 cu ft =) 49 lb of CO_2 gas, while face area of the hood (84 sq ft) requires (84 sq ft ÷ 1.4 sq ft =) 60 lb, or a total of 109 lb, therefore three 50 lb. cylinders of gas are required.

By NYFD rules (Fig. 1, Col. 2), same installation, only a total of 98 lb required, thus 777 cu ft ÷ 8 cu ft = 98 lb, therefore, only two 50 lb cylinders are required.

Comparison in the example is on the fire department basis of flooding only the hood and the duct up to the automatic fire damper (Fig. 1, Col. 2 versus Col. 4). Where, however, the 25 ft of duct run from the range to exhaust riser must also be protected, as required by NFPA rules (Fig. 1, Col. 3), the additional duct volume of 438 cu ft calls for a further supply of gas to the amount of (438 cu ft ÷ 16 cu ft =) 27 lb. This gives a total of 136 lb, so that three 50 lb cylinders must be installed. Additional piping and nozzles between the damper and riser are also required.

Cost of system is affected more by the number of cylinders required than by the number of nozzles. Aside from extra nozzles to exhaust riser as required in the underwriters—approved CO_2 system, nozzles required will be about the same whether system is laid out by NFPA or NYFD rules.

*This means that the volume of space protected is the same. Rules for Col. 1-2 do not call for protection by screening face area of hood as in Col. 3-4-5.

might render nil the entire expenditure.

The cost of a CO_2 system is not a straight line upward with increase in length of ducts or range hoods, but increases rather as the requirements call for more gas cylinders. These cylinders are furnished in 50-lb units, so that a requirement of, say, 60 lb could be met only by furnishing two cylinders. Normally, at regular hourly rates, a one-cylinder underwriters job costs about $500, a two-cylinder installation about $800. These figures would be greater by about 20% where overtime rates are paid as usually required.

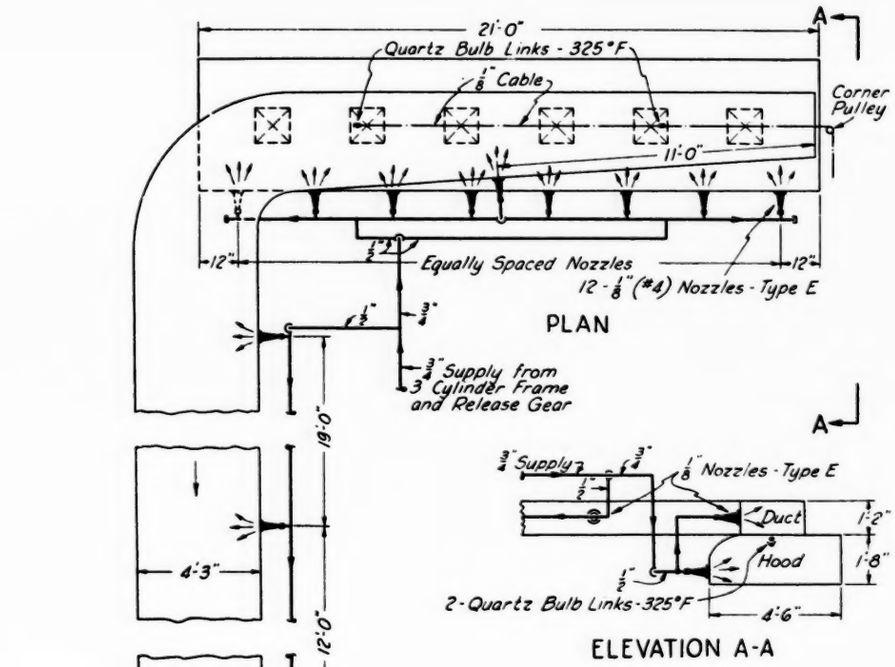
The exact cost to refill a gas cylinder depends upon the locality. Average cost is $6.50. Each cylinder is checked once a year by weighing to see that the weight does not go below 10% of the weight figure stamped on the unit.

In the case of the steam extinguishing system, one hotel estimates that it would cost them $7.00 per day just to maintain the pressure of 15 lb required by fire department rules. This includes no costs for maintenance of steam lines against leakage and for insulation repair.

Steam Smothering System

In the steam system, sprinkler heads are open and steam pressure at 15 lb minimum is always on up to the slow opening valve (Fig. 1 and 2). This valve should not be so close to the range that it cannot be reached at the time of a fire. The system, then, is manual, except for the automatic fire damper. This damper is actuated by the melting of a fusible link, and this link is at the damper rather than at the immediate source of the fire. For both of these reasons, the damper is slow acting compared to the pressure actuated damper of a CO_2 system.

While the fire department wants the CO_2 system to be both manual and automatic, and they are so arranged, they want the steam system to be manual only, because they cannot obtain it otherwise. There is no approved or sure method for automatic release of steam to outlets as there is with CO_2. Steam, of course, could be maintained up to closed sprinkler heads, though



Notes: Pipe and fittings conveying carbon dioxide to be ½ in. except as otherwise noted. Hang all pipe and conduit securely. No bends or offsets in control cable conduit.

Fig. 3. Underwriters approved CO_2 system for hotel kitchen range hood and duct.

steam at a temperature corresponding to 15 lb pressure would melt the solder links at the heads. In short, CO_2 is the preferable medium, but steam has been advocated for many years for the purpose in mind and there is no desire to penalize the owner who has a convenient steam supply by preventing him from using it.

Steam was at one time recommended by the underwriters in the absence of any other medium of a practical nature. The underwriters now say that steam to serve any useful purpose must be available in a most plentiful quantity at a temperature which will produce an air temperature of 175F. or the steam won't diffuse throughout the duct. Unfortunately, this is a temperature at which grease melts to spread a fire. Steam can be used as a maintenance proposition since it removes much of the grease, though not in a way cleanly to the surroundings. Incidentally, the flash point of grease varies from 480 to 580 F., except in bakeries where it may be as low as 390 F.

The fire department wants steam at a minimum pressure of 15 lb maintained at the manual operated valve 24 hours a day, "as long as the ducts remain in the building," to use their own phraseology. This is a large order. In questioning one of the fire department representatives he said he knew of no case where a system could be out of use, that they had yet to meet such a case. The writer can readily foresee such cases, for example, the Murray Hill Hotel. It was closed for wrecking, with all steam shut off, but the ducts still remained in the building at the time this matter was discussed.

Presumably, for much of the same reasoning, steam pressure should still be maintained as ruled when the kitchen hours are over, even though a range temporarily out of use cannot by itself set fire to the range hood or its connecting duct. The thought is that should there be a fire in the exhaust system after regular hours the watchman could open the control valve and find everything ready for action. With the kitchen out of use in after hours, the pressure might otherwise be reduced. In some cases this maintenance of pressure will require additional personnel and a special steam line (just for fire protection) with consequent operating costs. Is it any wonder, then, that in some installations where the ducts are short, the rules have been by-passed by removing the ducts and leaving the vapors to be removed only by an exhaust fan at the wall.

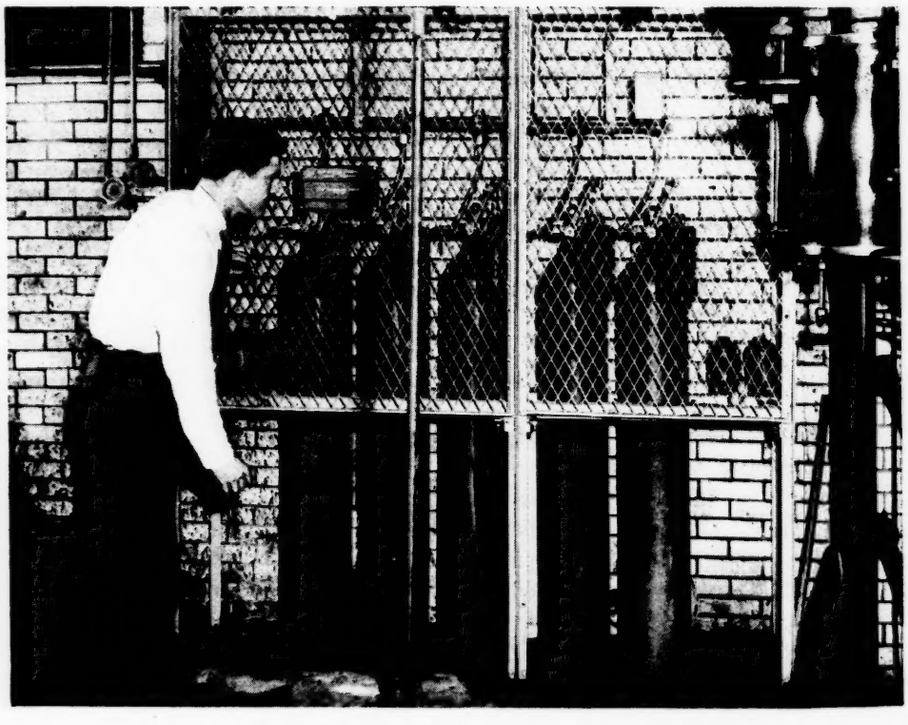


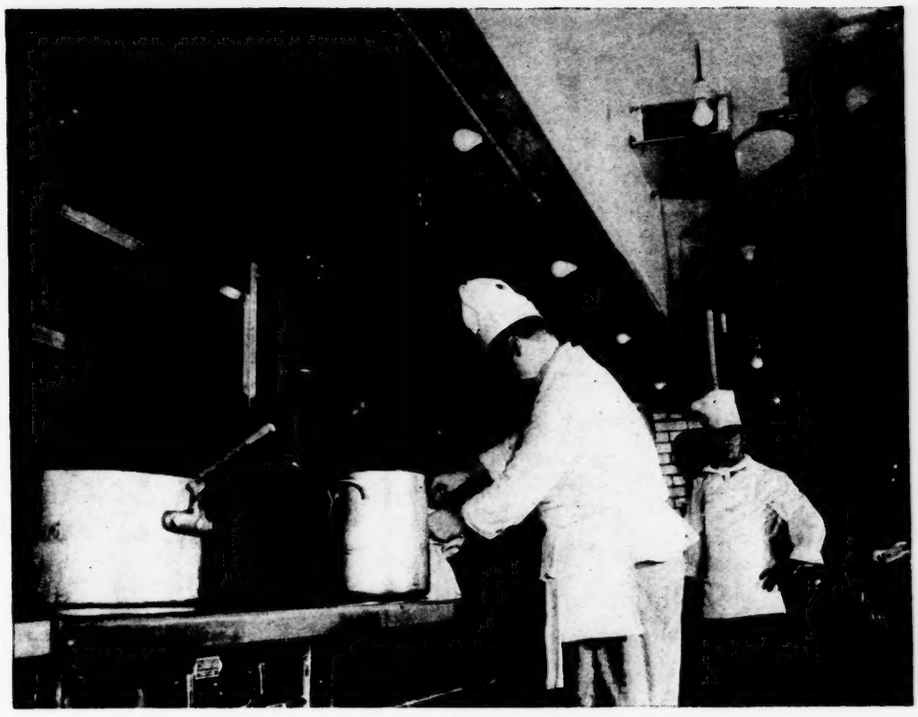
Fig. 4. Battery of cylinders for large CO_2 installation.

The reader may become intrigued by the reference in a previous paragraph to the slow opening valve. This manual valve is indicated on the sketches (Fig. 1 and 2) and must be so noted for the fire departments approval. It is not a special valve as one might be led to believe by its name wording, just a standard valve designated as slow opening by the fire department to prevent the use of a quick opening valve. Presumably, a quick opening valve, as commonly used in the past, might create destructive water hammer, but slow opening does not mean slow delivery to a system whose protection is based mainly on time in seconds.

In the case of the steam system, the fire department does not require automatic stopping of the fan at the time of a fire. Assurance may be felt that the automatic fire damper will close so quickly that the fan cannot pull air through the system to the extent of maintaining a fire. On the other hand, accumulated grease has been known to keep a damper partially or fully open. In the final analysis, automatic control of the fan is not easily accomplished in a steam system, and the system being manual, the stopping of the fan might as well be manual also.

The fire department does not find remote control of the fan necessary for a CO_2 system and their rules call for such control. In this case, the system is automatic, therefore, automatic remote control of the fan is a simple matter but such automatic control should be as necessary in one type of system as in the other.

Reference has been made previously to the automatic fire damper being rather close to the hood in a steam system. Under fire department rulings, the damper can be near or far. On contract jobs it is near so that less sprinkler heads will be required, therefore less cost. The fire department, however, prefer that the fire be confined to a minimum area where they can get at it. They don't want the fire to spread. Anything that retards the fire pending their arrival is to their advantage. Prevention of loss of life is their main consideration, property damage being left as the concern of the underwriters.



F.g. 5. CO_2 system blanketing face area of hood.

It is this line of reasoning, plus avoidance of hardship to some owners that leads to the difference in rules for CO_2 and steam systems, to some fallacies and half-way measures and a degree of protection in the case of a steam system that is most dearly bought. A steam system, even when cheapest in first cost, may offer insufficient or no protection at all and therefore be the most expensive in the end.

Advantages of CO_2 Systems

Carbon dioxide is the logical medium for extinguishing fires. The gas penetrates into every crevice and does not depend upon any source exterior to the premises or used for other purposes. A CO_2 system, being self contained, is ready for instant use to automatically, quickly and noiselessly extinguish a fire without damage to equipment and food. It cannot, as is possible with a steam system, create confusion by fog nor damage from greasy water.

Carbon dioxide is non-toxic and non-rusting and does not freeze or deteriorate with age. It can be safely used on live electrical equipment. One of its best demonstrations of harmlessness to food is given by the salesman as his meal is served in a restaurant. With a small portable CO_2 extinguisher he sprays his meal and then eats the food as though it were improved in quality.

Operation of CO_2 System

In a CO_2 system the gas is stored as a liquid under pressure of about 850 lb at 70F in one or more steel cylinders. (See Fig. 4). Each cylinder contains a 50-lb charge and weighs 165 lb when fully charged. When the gas is automatically or manually released it expands 450 times its stored volume to flood the space through small pipes to open nozzles which fill the ducts and blanket the face area of the hood with gas. (See Fig. 5 and 6). The air content of the protected space is diluted and its contained oxygen of normal 21% thereby reduced to about 15% at which point fire cannot exist beyond a few seconds. The pressure of the gas is also used to operate devices which stop the exhaust fan, close the fire damper and shut off the gas supply to the range.

Cylinders are 51 in. high, 8½ in. in diameter, with a required overhead clearance of about 15 in. Area required for cylinders and rack is about 12 in. square for one cylinder, 19 in. x 15 in. for two cylinders and 52 in. x 18 in. for three cylinders. Piping is galvanized steel with extra heavy malleable iron fittings. The control cable is 1/8 in. run in 3/8 in. galvanized conduit using 90 degree corner pulleys at changes in

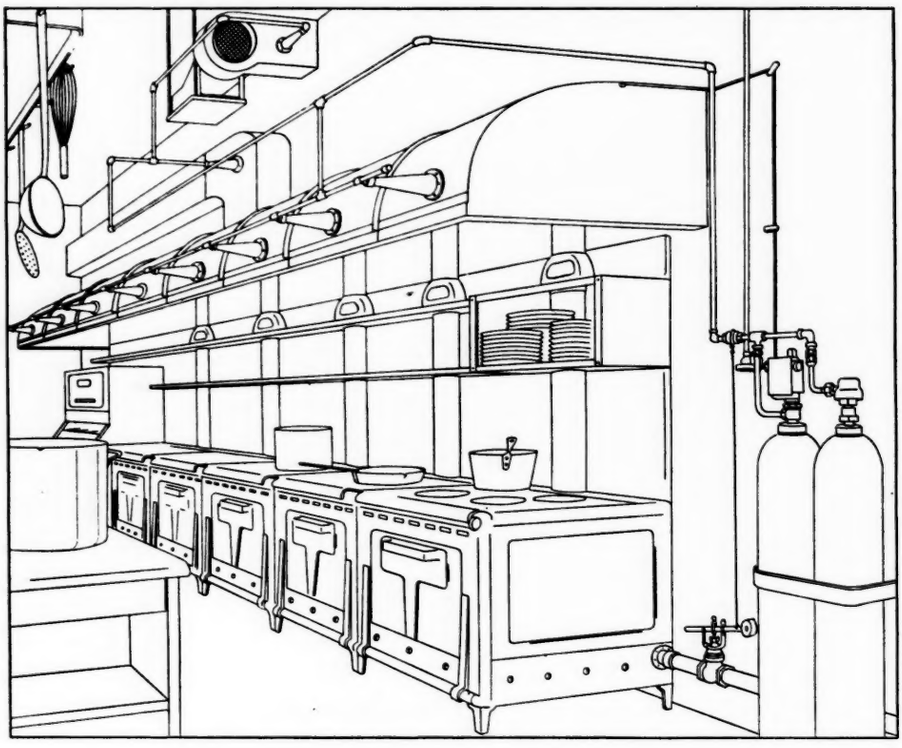


Fig. 6. A typical CO_2 system for kitchen range hoods.

direction. If the cylinder location is liable to be one with a temperature above 150F, such location should be well ventilated.

Dampers in General

As a rule the outdoor opening to an air conditioning fan is automatically closed off by means of louver dampers with the stopping of the fan. In the case of hood exhaust from kitchens, the underwriters, however, want no dampers within the ducts, except for CO_2 systems where one has to confine the gas and the automatic fire damper operating mechanism is most positive in action and the damper remote from outdoors. Systems not protected by CO_2—smoke, or are liable to do so, therefore the underwriters want the smoke to go out of the building by the flue effect of a fire within the ducts and the fan to be stopped at such a time so as not to increase the fire.

The experience of pneumatic and of sheet metal workers is that grease will congeal on dampers in such ducts, so that dampers soon stop operating, and further, that after outdoor dampers are closed in freezing weather a blow torch is sometimes required to thaw them out. Notwithstanding such experiences, dampers will occasionally be installed in such places by those who are fearful that the kitchen might otherwise freeze up when the fan is not running in non-working hours.

A particular illustration of the point in question is shown by the accompanying sketch (Fig. 7), an actual installation where outdoor temperatures sometimes drop to —30F. Of course, with such a low temperature and a total outdoor louver area of 50 sq ft, it is natural that some consideration should be given to the possibility of freezing up in the kitchen.

What, then, are such possibilities? In the first place, the area of the outdoor louvers need not be considered so much as the 30-sq-ft area of the ducts. In other words, the louvers can pass no more air backwards than the area of the ducts will allow. The force of the wind must also overcome the resistance of the louver, large fans, the ducts with their turns, the stack or flue effect tendency of the ducts to opposing flow, and finally the considerable resistance of the filters in the hoods.

In the case shown, the louvers and the kitchen are located on the same exposure. Hence, there will be infiltration of outdoor air through the window cracks to the kitchen. This will raise the pressure in the kitchen to a slight extent and to that extent oppose any backward flow of air within the ducts.

It is a simple matter to maintain sufficient heat against freezing, or to close the slide dampers at hoods. This should be done in any event, so as to reduce heat loss from building construction and any radiators in which heat is maintained. The residual heat from construction,

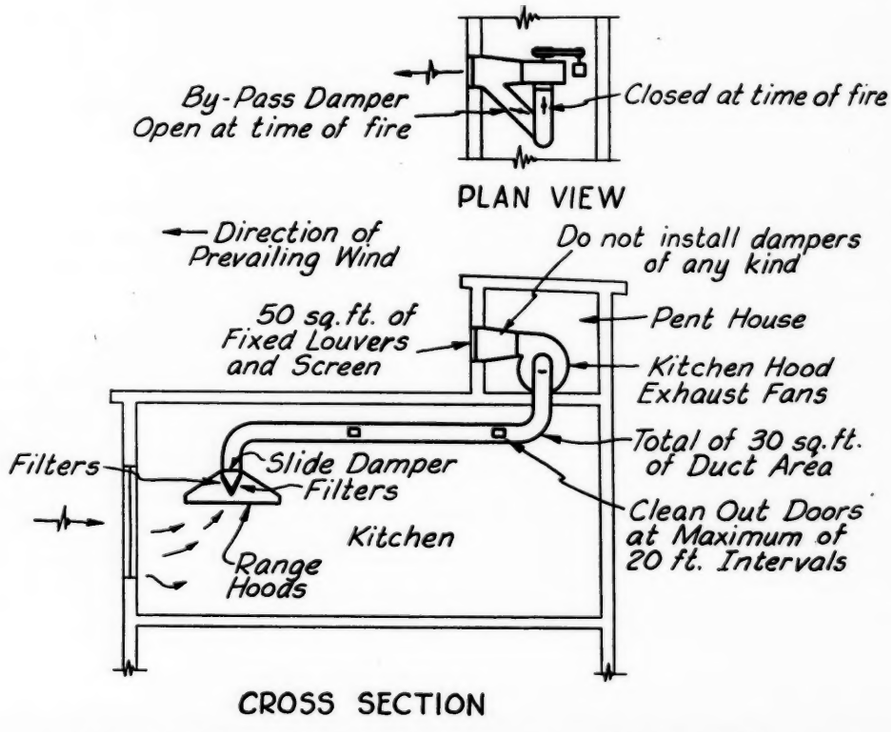


Fig. 7. An installation where damper was proposed to prevent freezing.

Fig. 8. Automatic closure gas valve on CO_2 system.

furnishings, adjacent floors and partitions and water within equipment and service piping will keep the usual kitchen from freezing. The larger the louver area, the larger the kitchen will be, and the less likelihood of a freeze-up.

Some years back we were accustomed to design with an automatic by-pass around the fan as shown in plan view of the sketch (Fig. 7). This would prevent the fan from being damaged by fire, providing the dampers worked. The fan, however, can best be protected by remote control of its motor from the kitchen at some point near the hood. Sometimes thermally operated units are located in the hood for the same purpose. On the other hand, a photo electric eye in the duct to supersede any normal damper control (so as to keep the damper open when there is a fire and the fan is stopped) would soon grease up even if the damper did not.

The writer in contacting an insurance representative out West was surprised to learn that he wanted the installation of automatic dampers at the fan discharge of a kitchen exhaust installation in his locality (Fig. 7). His reasoning was based on the premise that a range hood fire might spread to contact the pent house wall in which the fan discharge louvers were located. For real fire protection he recommended a vertical duct riser, preferably outside of the building, but otherwise inside within a shaft of 4-in. hollow tile, the riser to terminate well above the roof with an inverted sprinkler head at the top. An access door at the base of the riser would provide for the occasional cleaning out of any condensation. A steam nozzle at the base of the riser was also proposed, one that could be throttled down to pass enough steam for continuous cleansing of the riser, but at the time of a fire opened wide for smothering the fire.

Grease Filters

A further recommendation was that the grease be intercepted and collected at its source by using water cooled filters in the kitchen range hoods. The writer has been unable to locate anyone who makes such a filter, and in any event is still wondering what it would do other than to add resistance to the fan, and, if efficient, make possible a more lively fire at the filters. Furthermore, the greasy vapors would be so diluted with air and steam from the range that the cooling filters would have to be of unreasonable size and depth and even then be capable of lowering but little of the vapors to the dew point. Incidentally, one user of grease filters makes his own from a series of chains which he removes from the range hood each night and dips in carbon tetrachloride for cleansing. Let us hope that he always remembers to return them to their proper place each morning.

The writer has observed many ingenious methods of preventing or putting out fires in kitchen exhaust systems. One department store uses a powdery substance in the ducts to absorb the grease which accumulations can then be removed at proper intervals, but insurance companies have not ruled on this sort of thing. They do suggest the use of hand extinguishers of the carbon dioxide or liquid vaporizing type for small systems and for large systems the piping of carbon dioxide under thermostatic control to the ducts as previously described herein. One insurance representative suggested the use of air conditioning nozzles or automatic sprinklers to spray water within the ducts. Such a method would most certainly subtract a good part of the ceiling overhead in the event of a fire.

In general, insurance requirements of today are most sensible. Their rulings depend a lot upon the kitchen management as to good housekeeping and the keeping of grease filters in place at the hoods when the ranges are in use. As the filters get loaded, they cause a noticeable reduction in draft. They should, however, be cleaned once or twice a week regardless of loading and this can be done with a strong washing powder and steam. The filtering media should be of a type to resist passage of a flame and of sufficient coarseness to stand up under a fire without collapsing, melting down, or burning with a shower of sparks.

Good Housekeeping

Washing or swabbing of the ducts with a non-flammable grease solvent such as carbon tetrachloride is effective, but expensive, unless the system is small. Access doors should be substantial, tightly fitted, strategically located at turns and not more than 20 ft, apart on straight runs. They should, where practical, be taken from the side of the ducts to prevent dripping of residue. Drip pans should be installed below cleanouts when these are in the bottom of the ducts. Fans should be provided with asbestos sleeve connections at inlets and outlets, but try to obtain one that doesn't leak air.

All hoods should be insulated with 2 in. of 85% magnesia, finished with 1/2 in. hard cement where exposed. Ducts and fans require at least 1 in. of 85% magnesia. Underwriters rules call for sheet metal of ducts from hoods over cooking equipment to be a minimum of No. 18 U. S. Standard Gage, but No. 12 gage black iron is preferable for ducts from range hoods and bakeries and No. 16 galvanized iron for ducts from dish and glass washers. Ducts should be constructed with angle iron corners and tight seams.

Open to Debate

One insurance requirement that does not seem justified is the stipulation of air velocities within the ducts at a minimum of 2000 fpm. Presumably, this is required to keep ducts free from grease and to settle out the grease in a trap with cleanout at the base of the exhaust riser,

the riser being usually assumed as outdoors. Such velocities make for noise and excessive friction.

Another most debatable requirement is that the inside laps in duct joints project in a direction against the air flow. This method adds friction, collects grease at the laps and creates possible drip points. No doubt the theory is that most of the ducts are vertical (which they are not as a rule except in tall hotels), therefore the descending drip would not be caught by the laps. Assuming that a riser does not drain itself free of all grease, just how could it be swabbed or scraped from the cleanout at the base of the riser when working against the laps. Horizontal ducts could be inclined to a grease trap, though an inclination sufficient for good drainage might not look so well even if it could be obtained.

SYSTEM REQUIREMENTS

New York Fire Department Requirements for Fire Extinguishing Systems in Hotel Kitchen Ranges and Their Exhaust Systems

Steam Extinguishing Systems – January 3, 1947

(1) Hood and exhaust ducts thoroughly cleaned and in good repair before system is installed.

(2) System manual in operation. Provide source of steam from high pressure boiler or other approved source equal to at least 15 lb pressure.

(3) Steam pipe size at source of supply, not less than 1½ in., and decreased 3/4 in. at most remote steam jet or outlet which shall be standard 1/2-in. open sprinkler heads located not more than 8 ft on centers within the duct area with additional head located 2 ft inside the duct opening from hood.

(4) Pipe, fittings, valves, steam, standard. Control valve sealed open located at source of steam supply, also a slow opening control valve sealed closed located on steam extinguishing line outside cooking range area or at least 10 ft from range.

(5) Control valves located in accessible position for prompt operation sealed in approved position with light wire seal. Tag to indicate purpose of valve.

(6) Paper sketch of proposed system filed with department for approval before any work is started. Show or state all details necessary to comply with these rules.

(7) Source of steam supply designated (boiler or street supply) must be charged with steam at required pressure continuously or as long as the ducts remain in the building.

(8) Steam trap installed on steam line in such a position to remove condensation up to the valve located at or near operating valve at ducts.

(9) Names of employees instructed in operation of system posted at or near operating valve at ducts.

(10) Rules are general. May be changed in part to meet requirements of construction, layout, existing equipment, and location of valves. Such changes show in detail on approved plan.

(11) In case an adequate supply of steam at the required pressure is not available, a proper supply and arrangement of CO_2 extinguishers will be accepted.

Author's note: Referring to last part of (4), the wording "at least 10 ft from range" means that control valve can be some distance from range, but cannot be closer than 10 ft.

An approved fire damper is not called for in rules, but must be installed and shown on sketch for approval. It may be located close to range with a sprinkler head within 4 ft of hood side of damper.

Remote control of the fan is not required as called for in their rules for CO_2 systems.

Automatic closing gas valve is not required.

Rule (3) is not clear. It means that a head must be located 2 ft up inside each uptake as shown Fig. 1 and 2. Where duct is not directly over range hood, as in Fig. 1, additional line of heads must be located in duct on not more than 8-ft centers.

CO_2 Extinguishing System – November 1, 1946

(1) Hood and exhaust ducts thoroughly cleaned and in good repair before the system is installed.

(2) Must be manual and automatic in operation.

(3) Entire hood and each duct, also any deep frying units provided with adequate number of nozzles; one nozzle at inlet of each duct and one at inlet of damper. Duct nozzles spaced not more than 20 ft apart.

(4) Automatic operation arranged to cut out exhaust fan and close damper or doors in the exhaust system. Automatic releasing equipment of standard design and heat detecting units properly located.

(5) At least one cylinder with 50 lb of gas provided for each cooking range installation up to 400 cu ft of hood and duct area to the damper, with an additional amount of gas and in not less than 50-lb cylinders for each additional 400 cu ft, or portion of such area.

(6) Paper or cloth prints of above system filed with department for approval before any work is started. Show all details as to hoods, ducts, dampers, heat detector units, pipe, nozzles, gas cylinders, automatic trip, manual operation, etc.

Author's note: Obviously, in (5), the word "area" should be changed to "volume" since there is no such thing as cubic feet of area.

Underwriters (NFPA) Fire Protection Requirements for CO_2 Systems Protecting Kitchen Hoods and Ducts

(1) Deep fat frying unit, 1 lb gas for each 0.6 sq ft of liquid surface. Example: 2 ft x 3 ft unit requires 10 lb CO_2 gas.

(2) Bottom opening of hood requires screening gas at rate of 1 lb of CO_2 for each 1.4 sq ft of area. Example: Hood 3 ft wide by 14 ft long requires 30 lb gas.

(3) Cubic volume of hood plus exhaust ducts up to damper requires CO_2 at rate of 1 lb for each 16 cu ft of volume. Example: Hood 3 ft wide by 2 ft high by 14 ft long with duct 12 ft x 12 ft x 40 ft to damper requires, 8 lb of gas.

(4) All these, when added, give a total gas requirement of 48 lb or one 50-lb cylinder for an underwriters-approved system. By NYFD rules, 1 lb per 8 cu ft hood only or 10.6 lb total gas requirement. (No screening required).

Other Requirements:

(1) Damper in exhaust duct weight operated and provided with a gas (CO_2) operated pressure trip for automatic closing upon discharge of CO_2 system.

(2) Pressure switch (CO_2 operated) provided to shut down exhaust fan upon discharge of system.

(3) Belfield gas valve installed in cooking gas supply line for automatic closure.

Author's note: Screening of face area of the hood is called for in (2) but is not required in fire department rules.

HEATING AND VENTILATING'S
Reference Section

Condensation of Refrigerants on Finned Tubes

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Professor of Chemical Engineering, University of Michigan

Condensation of Refrigerants on Finned Tubes

D. B. ROBINSON and D. L. KATZ

Wolverine Trufin Fellow, and Professor of Chemical Engineering, respectively, University of Michigan

Data on the condensation of refrigerants on finned tubes are reviewed and methods for predicting the condenser area for a given heat load are presented. The effects of non-condensables, superheat, tubes in a vertical row, and fin efficiency are discussed. Condensing film coefficients have been calculated for propane, methyl chloride, Freon 11, 12, and 22 and water-side coefficients for tubes of three different diameters. Use of the overall coefficients, either experimental or calculated, is illustrated in an example.

THE purpose of this article is to present some of the information that has been gathered on the use of integral spiral finned tubes for condensing refrigerant vapors.

Extended Surfaces

Extended surfaces have been successfully applied for several years where it has been necessary to improve rates of heat transfer. Perhaps the oldest and most essential applications are found where heat is being transferred to air. The extended surface, as fins of various types, increases the surface area in contact with the air to 10 or 20 times that of the tube and generally increases the heat transfer in proportion to the surface added.

The rates of heat transfer for air heaters are low, about 5 to 25 Btu per hr sq ft F. Heat transfer rates in condensers are usually well over 100 Btu per hr sq ft F, and the advantages of extended surfaces decrease for these higher rates of transfer. Mechanical difficulties in fabricating condensers having extended surfaces have also limited the extension of extended surface tubing to condenser applications in years past.

Recently, a process has been developed whereby the extended or finned surface is extruded from the surface of a thick-walled plain tube for making low integral spiral finned tubes having a ratio of outside to inside area of about 3.5 to 1. The plain ends of the tube have a slightly larger diameter than the finned section so that tubes fit standard condenser sheets. This feature makes possible the application of extended surface tubing in standard condensing apparatus. The type of tube is illustrated in Fig. 1. The ratio of the areas of finned and plain tubes fitting the same tube sheet is about 2.5 to 1.

An increase in outside area is most to be desired when the resistance to heat transfer is practically all on the outside. Refrigerant condensers often have more than 50 to 80% of the resistance to heat transfer on the outside condensing surface. However, even when the resistance on the hot side is somewhat less than that on the cold side, experimental and calculated information indicates that tubes having extended surfaces of the type mentioned possess significant advantages over the usual plain tubes.

Heat Transfer Calculations

Equation (1) expresses the fundamental relation involved in making heat transfer calculations, namely,

$$q = UA\Delta T \qquad (1)$$

where

q = heat transferred, Btu per hour

A = area in square feet

ΔT = temperature difference, F.

U = overall heat transfer coefficient based on area A, Btu per (hr) (sq ft) (F)

Symbols used in all equations are listed in a table of nomenclature, page 93.

If $\frac{1}{UA}$ is looked upon as an overall resistance, Equation (1) may be written in the form

$$q = \frac{\Delta T}{1/UA} = \frac{\text{driving force}}{\text{resistance}} \qquad (2)$$

As in electrical conductivity, resistances to heat transfer in series are additive. In condensation, where heat is transferred through a condensate film, a clean tube wall, and a water layer the total resistance becomes the sum of the three individual resistances:

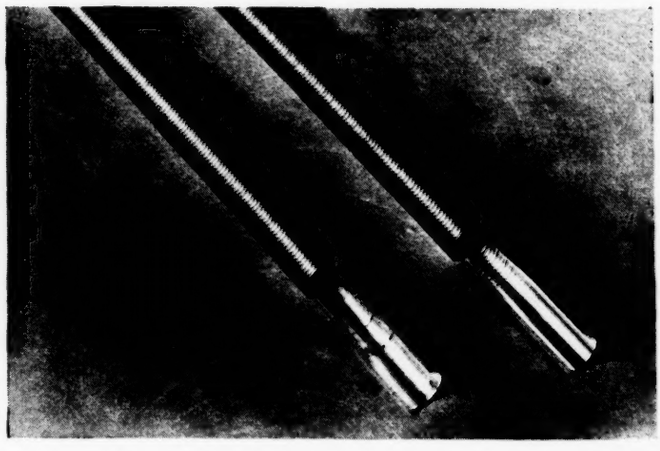


Fig. 1. Finned tubes showing plain ends.

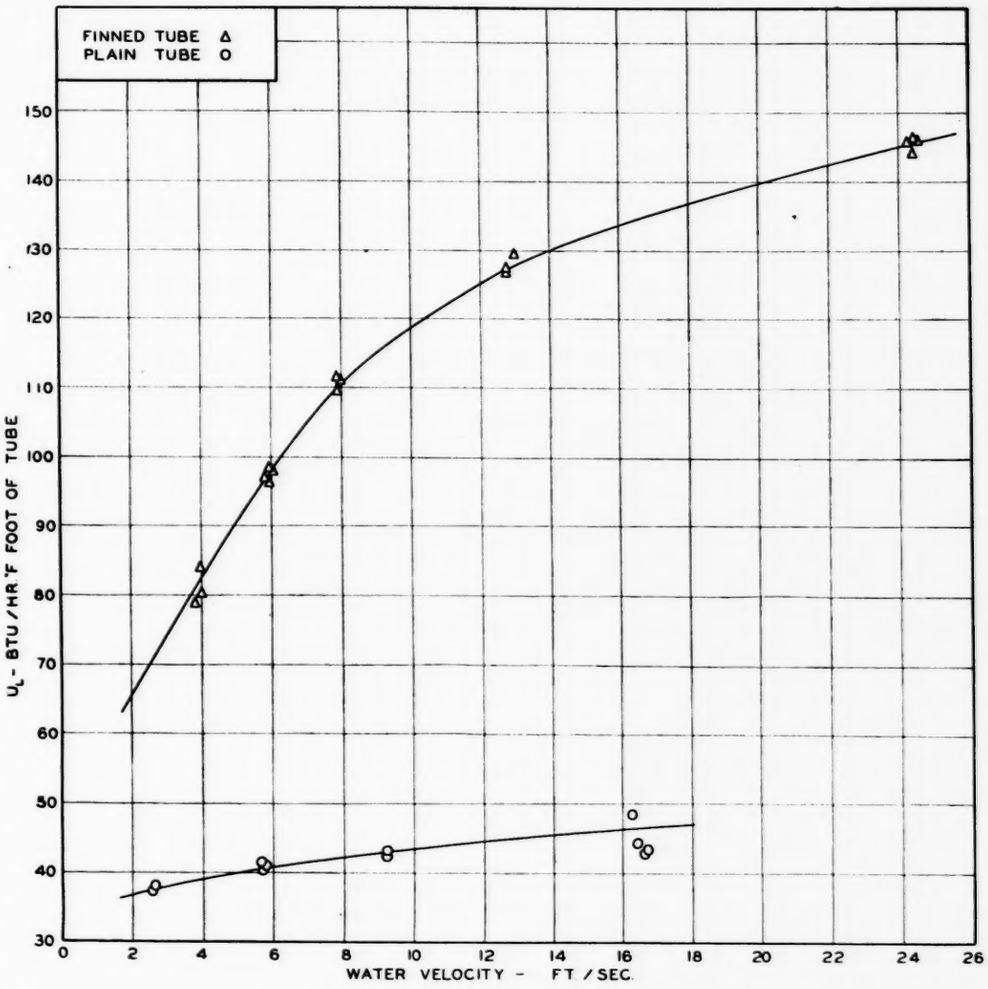


Fig. 2. Overall heat transfer coefficients per foot of tube versus water velocity for condensing Freon 22.

$$\frac{1}{U_o A_o} = \frac{1}{h_w A_1} + \frac{l}{k A_m} + \frac{1}{h_o A_o} \qquad (3)$$

That is:

$$\begin{pmatrix}\text{overall}\\ \text{resistance}\end{pmatrix} = \begin{pmatrix}\text{water}\\ \text{side}\end{pmatrix} + (\text{metal}) + \begin{pmatrix}\text{condensing}\\ \text{film}\end{pmatrix}$$

where

h_w = heat transfer coefficient on the water side, Btu per (hr) (sq ft) (F)
A_1 = inside tube area, square feet
h_o = heat transfer coefficient through condensate layer Btu per (hr) (sq ft) (F)
A_o = outside tube area, square feet
l = thickness of tube wall, feet
k = thermal conductivity of tube, (Btu) (ft) per (hr) (sq ft) (F)
A_m = average tube wall area, square feet

In the case of condensation on plain copper tubing, $1/kA_m$ is relatively insignificant and may be neglected. It is then obvious that an increase in outside area, A_o, will tend to reduce the overall resistance, particularly if the outside coefficient is less than the water side coefficient.

Horizontal tube and shell condensers are commercially important for condensing refrigerants. The lower the condensing coefficient, the greater is the advantage in using finned tubes for condensing organic vapors. When resistance offered by the condensing film is 80% of the total, the overall resistance to heat transfer will be decreased to about 52% of its value if the plain tubes are replaced by finned tubes having 2.5 times the outside area. In other words, only 52% as much finned tubing is required for the same job. In case the outside and inside resistances are equal, the tube has 70% as much resistance, and the length of finned tubing required will be 70% per cent as much as the length of plain tubing.

Fig. 2 is a comparison of heat transfer rates obtained for condensing Freon 22 on the outside of single horizontal tubes. The finned tube was 5/8 in. root diameter with 16 fins per inch 0.063 in. high and was condensing saturated vapors at 85F with an overall temperature difference between the vapor and the water of 36F. The plain tube was ¾ in. outside diameter and condensed Freon 22 under the same conditions.

The overall coefficients are based on the total outside area *per foot of tube* rather than on a square foot of outside surface, and are calculated from the equation

$$q = U_1 A_1 \Delta T_{oa} \qquad (4)$$

where A_1 is the outside area per foot of tube, U_1 is the overall coefficient based on that area, and ΔT_{oa} is the overall temperature drop. It may be seen that the finned tube transferred about 2.3 times as much heat

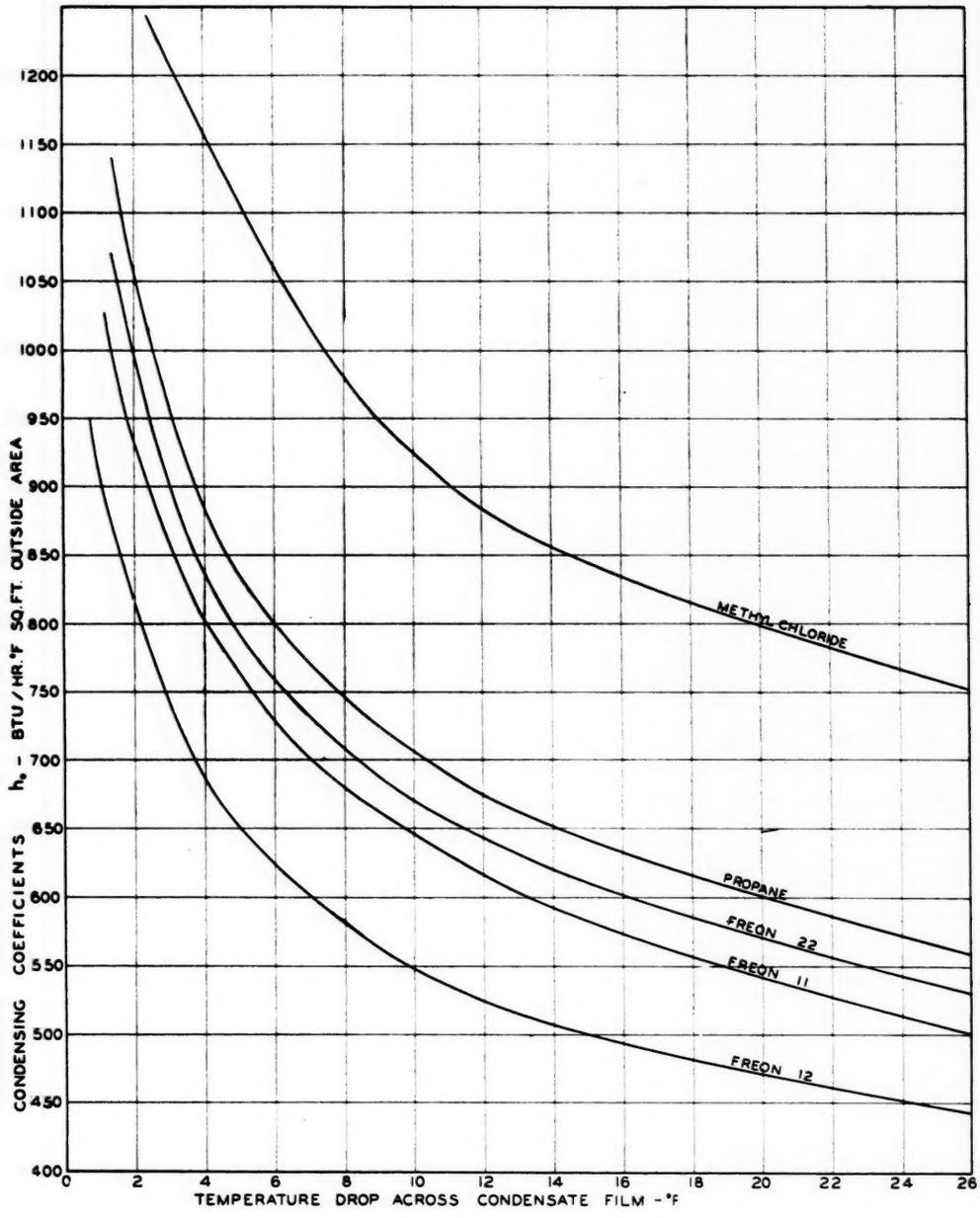


Fig. 3. Condensing film coefficients for a horizontal tubing having 16 fins per in. Condensing temperature 90F.

at a water velocity of 6 ft per sec. The ratio of the outside areas of the finned and plain tubes was 2.5.

When experimental data are available for overall coefficients, condenser design need not include calculation of the condensing film coefficients. In many cases, however, the conditions for a proposed condenser may be different from those for which experimental data are available. Then the condensing film coefficient and the water side coefficient must be computed separately and combined by Equation (3) to give a calculated overall coefficient.

It has been shown that Nusselt's theory can be applied successfully for the calculation of the condensing coefficients on finned tubes. The outside area is considered as the sum of a vertical surface and a horizontal surface. The resulting film coefficient becomes

$$h_o = \frac{1}{A_o} [\phi A_f h_v + A_t h_m] \qquad (5)$$

where

h_o = film coefficient for a single tube based on total outside area, Btu per (hr) (sq ft) (F)

A_o = total outside area, square feet

A_f = area of fins, square feet

A_t = area of plain tube having same diameter as the root diameter, square feet

h_v = film coefficient for vertical surfaces

$$= 0.943 \left(\frac{k^3 \rho^2 g \triangle H}{\mu \triangle T_v} \right)^{1/4} \left(\frac{1}{L} \right)^{1/4}$$

h_m = film coefficient for horizontal surfaces

$$= 0.725 \left(\frac{k^3 \rho^2 g \triangle H}{\mu \triangle T_v} \right)^{1/4} \left(\frac{1}{D} \right)^{1/4}$$

ϕ = efficiency of the vertical fin surface

From these definitions,

$$h_v = \left(\frac{D}{L}\right)^{1/4} \left(\frac{0.943}{0.725}\right) h_m \qquad (6)$$

Then from Equation (5),

$$h_o = \left[\left(\frac{D}{L}\right)^{1/4} \left(\frac{0.943}{0.725}\right) \frac{\phi A_f}{A_o} + \frac{A_t}{A_o}\right] h_m \qquad (7)$$

where

D = root diameter of tube, feet
L = average height of a fin, feet

Using these equations for methyl chloride, propane, Freon 11, 12, and 22, Fig. 3 has been calculated, assuming a condensing temperature of 90F. The same curves will be accurate to 5% for any temperature between 30 and 150F, for copper tubes with 5/8 in. root diameter, and 16 fins per in. each 0.063 in. high.

The heat transfer coefficient on the water side may be calculated by the method illustrated by McAdams. Curves shown in Fig. 4 were calculated for 5/8-, 3/4-, and 7/8-in. O.D. tube diameters measured over the plain end, and a water temperature of 75F.

Conditions Affecting Heat Transfer Coefficients

Non-Condensables

Performance of the shell and tube condensers is affected seriously by the presence of non-condensable gases such as air. Gases are in general relatively poor conductors of heat. If a vapor mixed with air strikes a cold surface, the vapor will condense and a film of condensate will form upon the surface. A surface layer of air will accumulate on this film, so that after condensation has commenced the cold surface will be protected in such a way that no further condensation will take place until fresh vapor displaces or diffuses through the air film. The controlling factor in the heat transmission then becomes the rate of diffusion of condensable vapor through a gas film, which is a slow process.

Othmer has shown that 0.5% air in condensing steam decreases the overall heat transfer coefficent to half its value with no air present, and 1.0% reduces it to one-third. In the course of experiments to determine heat transfer coefficients for butane condensing on a single horizontal finned tube, Beatty noticed that changes in overall temperature difference between saturated vapors and average water temperature for a particular series of runs did not produce changes corresponding to changes predicted by equations for condensing coefficients of heat transfer. After the apparatus had been thoroughly purged to remove non-condensables, heat transfer rates corresponded satisfactorily to the overall temperature difference. This indicates the disadvantages and misleading effects of non-condensables in condensing equipment.

Non-condensables should be reduced to the lowest possible value because they increase the resistance to heat transfer on the outside of the tube. If non-condensables are inevitable, an increase in external area would reduce the effect of the added resistance on the

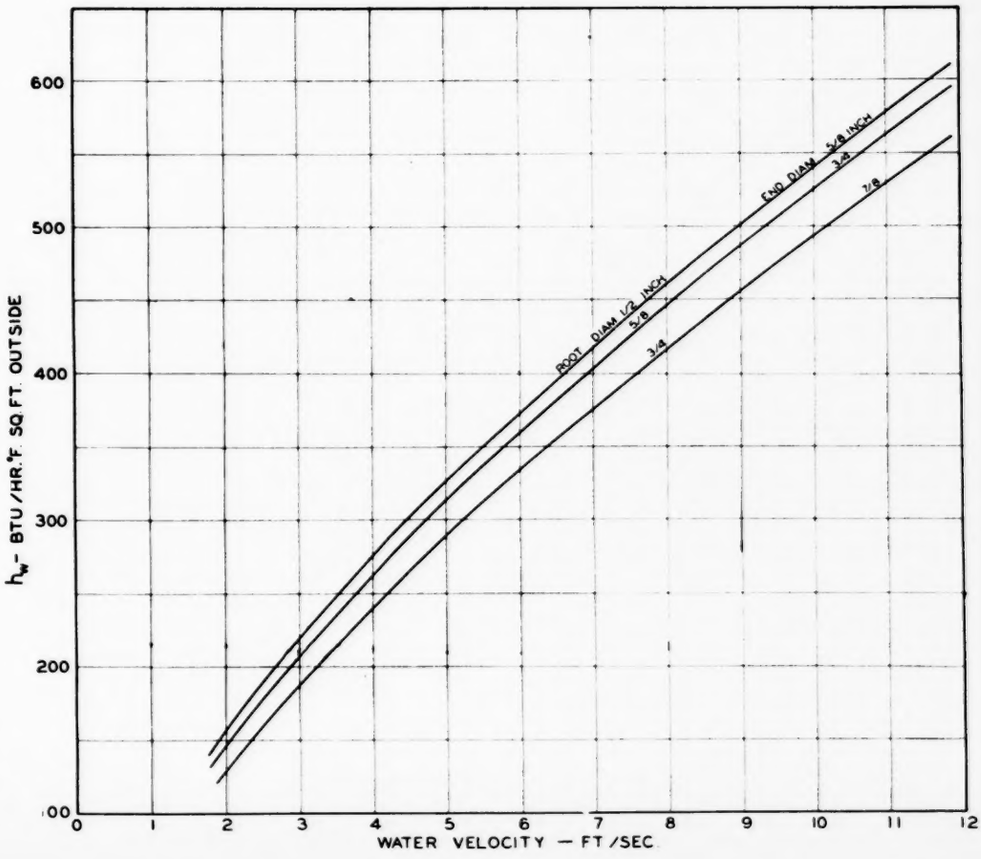


Fig. 4. Water-side coefficient calculated for finned tubes having the indicated end diameters and water temperature of 75F.

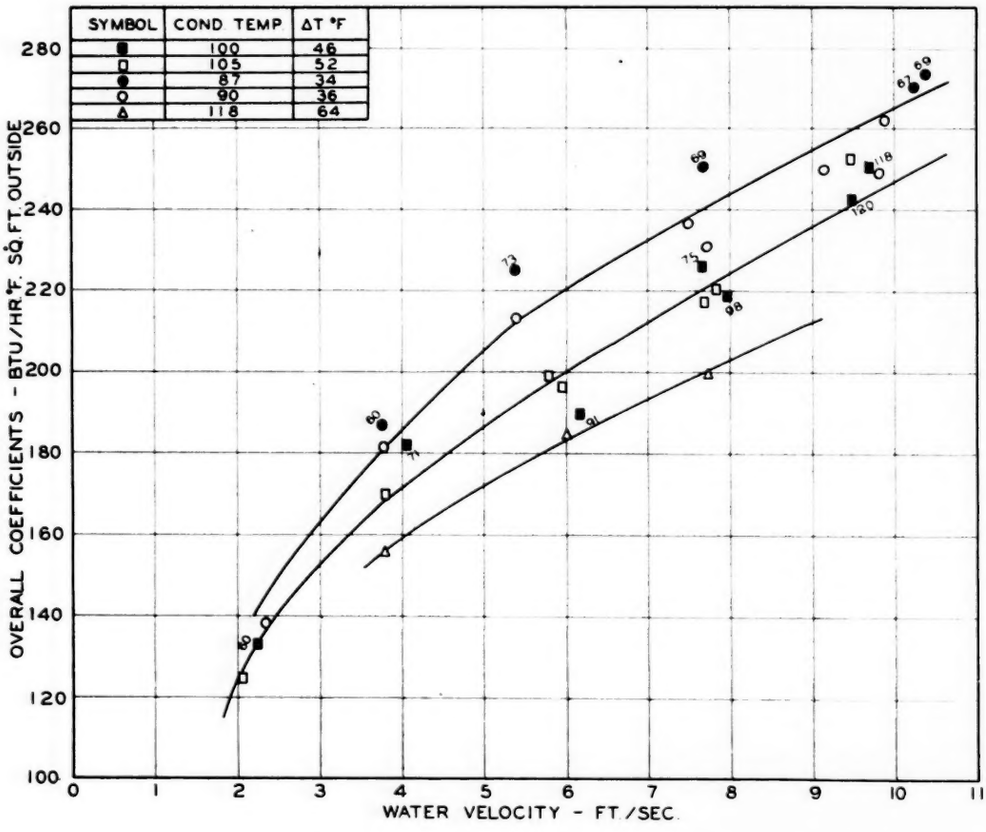


Fig. 5. Overall heat transfer coefficients for condensing superheated and saturated vapors of Freon 12. Numbers indicate degrees of superheat.

overall coefficient of heat transfer. Therefore, finned tubes would be an extra advantage in this case.

Systems of baffles are provided in condensers. These baffles are used to give a satisfactory flow pattern of the vapors over the tubes. In case a boiling range exists, or if truly non-condensables are present, such baffling is designed to insure that these components will be removed from the main portion of the condensing surface, leaving it free to condense the desired vapor.

Experiments have been run to determine experimental coefficients for condensing saturated and superheated Freon 12 vapors on a single horizontal finned tube. The saturated vapors were produced by a closed steam heated reboiler. They were condensed at the saturation temperature on the tube and then recycled. The superheated vapors were generated by a compressor on a refrigeration system and reached the condenser with 60 to 122F superheat. The curves in Fig. 5 present the experimental overall coefficients as a function of water velocity for a ¾-in. root diameter tube having 16 fins per inch 0.054 in. high. Coefficients for superheated vapors (solid points) and saturated vapors (hollow points) are essentially the same.

Heat transfer coefficients for condensing superheated vapors are calculated from the same equations that are used for saturated vapors. It is assumed that for a pure vapor no resistance to heat transfer exists between the vapor and surface of the condensate film. Therefore, the surface temperature of the film corresponds to the saturation temperature of the vapor at the condensing pressure. The value of ΔT used in Equation (1) will be the difference between this saturation temperature and the average water temperature. The overall coefficient will be the same as that for saturated vapor, but the heat load will be increased to allow for the superheat.

When the condensate from one tube falls to tubes beneath it in a condenser, the increased thickness of the film tends to reduce the rate of heat transfer. As the overall temperature difference is increased more vapor will condense and the layer of condensate on the tubes will become thicker. The increase in temperature drop across the vapor film reduces the coefficient of heat transfer in the film, so that the overall rate of heat transfer does not increase proportionally with the increase in overall temperature difference.

Experimental work on the condensation of vapors on six tubes in a vertical row indicates that the percentage decrease in heat transfer coefficient for tubes low in the row is small, with the fourth, fifth, and sixth rows being essentially the same. Fig. 6 illustrates overall coefficients obtained for the top tube, bottom tube, and the average of the six as a function of water velocity for condensing Freon 12 on finned tubes. The tubes were ⅝-in. root diameter and had 16 fins per inch 0.063 in. high.

Observation of the condensation showed considerable splashing and turbulence resulting from the swirling motion imparted by the spiral groove on the tubes. Such turbulence is considerably less on plain surfaces.

There are two reasons why the heat transfer coefficient does not continue to decrease proportionally to the distance from the top tube. Increased splashing resulting when the layer of condensate becomes increasingly thick actually limits the amount of condensate adhering to the tube to some value which is

apparently reached about the third tube. Also, the increased rate of condensation on the tubes changes the flow from viscous to turbulent motion, where heat transfer rates are increased and Nusselt's theory for streamline flow is no longer applicable without some modification.

When the properties of the condensing fluid differ from the properties of the fluid for which experimental coefficients are available, it becomes necessary to calculate condenser performance theoretically.

For multitube condensers, Merkel has proposed that the value of the film transfer coefficient calculated for single tubes by Nusselt's equation should be divided by $n^{1/4}$ where n is the number of tubes in a vertical row. A formula for finding the average number of tubes in a row for an ordinary shell and tube condenser has been derived. If n is the number of tubes in any row, then

$$n_{avg} = \left[\frac{n_1 + n_2 + n_3 + \ldots\ldots\ldots\ldots}{n_1^{0.75} + n_2^{0.75} + n_3^{0.75} + \ldots} \right]^4 \qquad (8)$$

where the subscripts refer to individual rows. Experimental results indicate that the value for n_{avg} should be modified, depending on the number of tubes in the row, so that the coefficient for a single tube could be divided by $(bn_{avg})^{0.25}$ when b is some number less than one.

When the total number of tubes in a bundle is greater than 25, Equation (8) can be closely represented by the following:

$$\text{For square pitch: } n_{avg} = 0.82\sqrt{(N)} \qquad (9)$$

$$\text{For triangular pitch: } n_{avg} = 0.40\sqrt{(N)} \qquad (10)$$

where N is the number of tubes in the bundle.

Fin Efficiency

The use of Nusselt's equations for calculating condensing film coefficients is only justified if the tube wall surface is all at the same temperature, or nearly so. In the case of a plain horizontal tube, the only differences in temperature will result from the changing thickness of the condensate film around the tube, and an average temperature can be used.

The tube wall temperature depends on the rate at which heat is being transferred and on the material of which the tube is constructed.

In the case of finned tubes, heat being transferred through the fin has a greater distance to travel than the heat transferred through the horizontal portion. There will therefore be a temperature gradient from the tip to the base of the fin. This temperature difference between the tip and the base will increase as the height of the fin increases, as the rate of heat transfer increases, and as the thermal conductivity of the metal decreases.

In condensing vapors, the tip of the fin will be at a higher temperature than the base and therefore the tip will be less effective than the base for condensing. To calculate condensing film coefficients for finned tubes using Nusselt's theory, the external area is divided into vertical and horizontal portions. Since the vertical or finned portion is less effective than the horizontal portion because of the temperature gradient in the fin, it must be multiplied by a factor denoted by fin efficiency before the corresponding effective external area is known. Fin efficiency has been defined by Gardner as

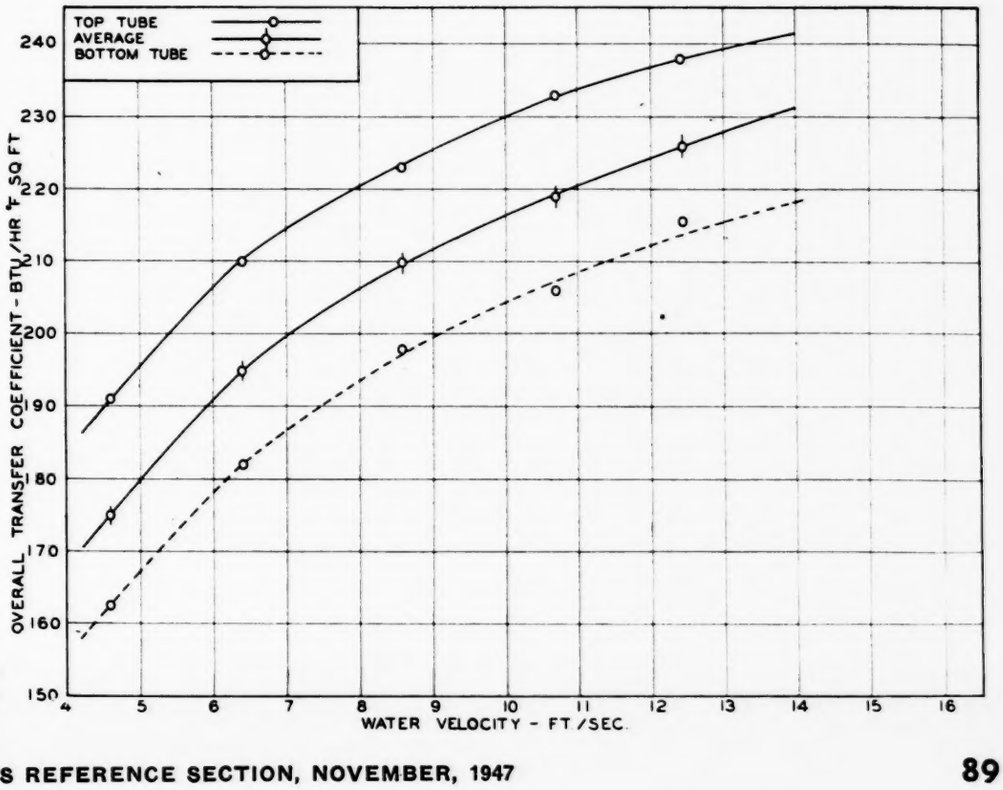


Fig. 6. Relative condensation rates of Freon 12 on 6 tubes in a vertical row.

$$\phi = \frac{A_{eff}}{A_t} \qquad (11)$$

where

ϕ = fin efficiency
A_{eff} = effective external vertical surface
A_t = total external vertical surface

The function has been calculated for a number of external shapes.

In Fig. 7, fin efficiency is plotted as a function of condensing film coefficient using thermal conductivity of the metal as a parameter for a low fin tube. The calculation is based on the curves presented by Gardner for annular fins with constant metal area. Dimensions of the tube for which these curves have been drawn are as follows:

Diameter over fins	0.750 in.
Root diameter	0.625 in.
Height of fin	0.063 in.
Thickness at tip	0.000 in.
Thickness at base	0.024 in.

For low finned copper tubes such as this one, at moderate rates of heat transfer the fin efficiency would only be slightly less than 100%. For low finned tubes constructed of steel or cupro-nickel, where the metal resistance is relatively high, the fin efficiency is lower. It decreases rapidly as the fin height increases.

Performance of Commercial Condensers

Experimental data for condensing refrigerants in multitube condensers using finned tubes are available from two sources.

The results of the experiments of Katz, et al, are plotted in Fig. 8 as rate of heat transfer versus water velocity. The data were taken while condensing Freon 12 in a 40-tube condenser using 5/8 in. root diameter tubes having 16 fins per in. 0.063 in. high. The condenser area was 65.6 sq ft. The experimental points are in general agreement with the curve but give higher rates of heat transfer than the calculated values. Overall coefficients were calculated for similar condensing conditions and water rates by equations based on Nusselt's theory of condensation modified for finned tubes and for number of tubes in a vertical row. The modification for tubes in a vertical row is based on

$$\text{Merkel's value, } h_{multitube} = \frac{h_{single}}{n^{0.25}} \qquad (12)$$

The second set of experimental data with finned tubes was reported by Jones and was taken while condensing Freon 11 in a commercial condenser having 185 tubes which were the same as those in the 40-tube condenser. The experimental and calculated rates of heat transfer are plotted versus water velocity in Fig. 9.

It is difficult to explain why these experimental data are not consistent with the theory. According to the data, the temperature drop across the refrigerant film is constant regardless of loading. Also, the highest values for refrigerant film coefficients were obtained when the overall temperature difference was greatest.

The rate of heat transfer at various water velocities was calculated for a constant heat load of 3,000,000 Btu per hr and a condensing temperature of 90F. The calculated rates of heat transfer are plotted in Fig. 9 for a single tube and using the actual number of tubes in a row. It will be noticed that the experimental rates of heat transfer reach the calculated rates when the heat load is very high. The low rates of heat transfer at low loads can be explained only on the basis that possibly non-condensables were present. At high loads

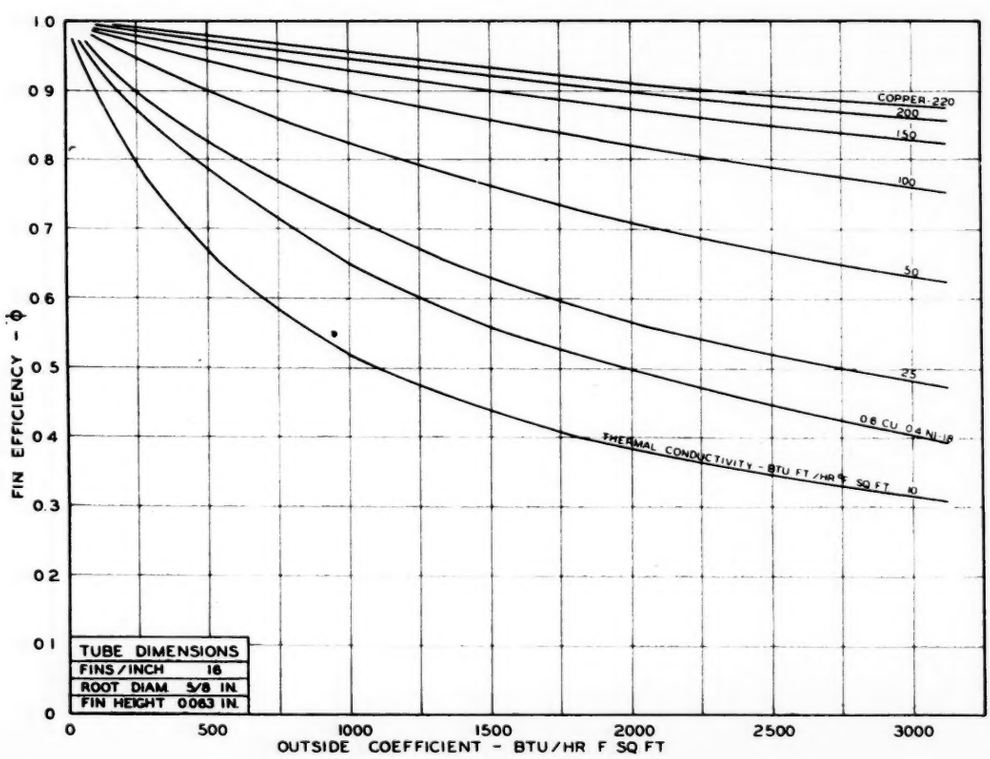


Fig. 7. Fin efficiency as a function of outside coefficient and thermal conductivity.

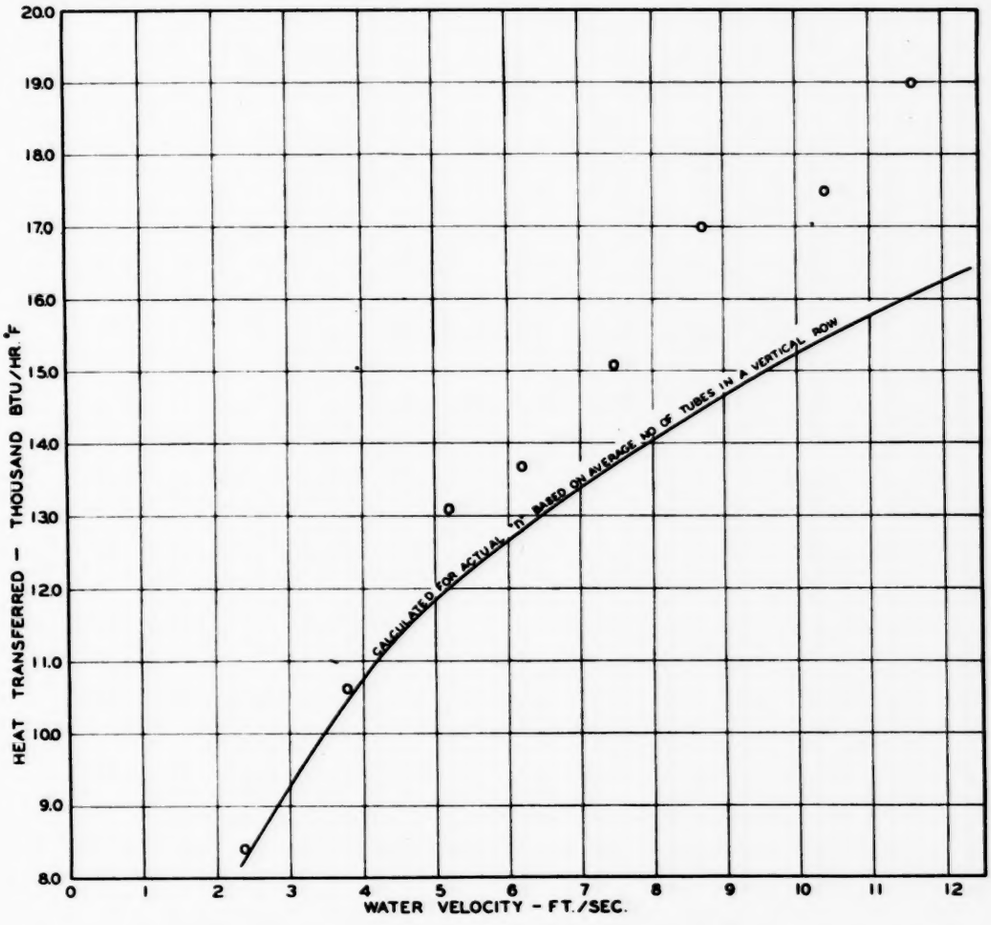


Fig. 8. Experimental and calculated rates of heat transfer for condensing Freon 12 in a 40-tube condenser having 65.6 sq ft area.

and corresponding vapor velocities, the bulk of the heat transfer surface became effective and gave coefficients in accord with predicted values.

Example of Condenser Design

Problem: Calculate the condensing surface required to condense 14,140 lb per hr of Freon 12 which will enter the condenser at 140 lb per sq in. absolute and 124F. Cooling water is available at 65F. Low finned copper tubes 5/8 in. root diameter, 3/4 in. ends, having 16 fins per inch are to be used. The I.D. of the tube is 0.545 in., the outside area is 0.508 sq ft per ft, and the ratio of the outside area to inside area is 3.56.

Method 1. Using an Overall Coefficient from Multitube Experiment

(1) Heat in entering vapor above —40F = 92.3 Btu per lb.

(2) Heat in condensate at saturation temperature, 104F ..... = 32.5 Btu per lb.

Heat lost in condensing (1)–(2) ..... = 59.8 Btu per lb.

Therefore, total heat load = 59.8 (14,140) ..... = 846,000 Btu per hr. or 235 Btu per sec.

Assume a water velocity of 6 ft per sec.

Then from Fig. 8, heat transfer vs. water velocity,

$$U \text{ (experimental)} = \frac{13{,}850}{65.6} = 211 \text{ Btu per (hr) (sq ft) (F)}$$

From the dimensions of the tube:

Outside area ..... = 0.508 sq ft per ft

Inside diameter ..... = 0.545 in.

Assume a water temperature rise of 15F

Then water rate = heat load in Btu per sec / water temp. rise $= \frac{235}{15} = 15.7$ lb per sec or 0.252 cu ft per sec

$$\text{Area of tube} = \frac{\pi}{4}\left(\frac{0.545}{12}\right)^2 = 0.00162 \text{ sq ft}$$

$$\text{Total area for water required} = \frac{0.252}{6} = 0.042 \text{ sq ft}$$

$$\text{Then number of tubes required} = \frac{0.042}{0.00162} = 26 \text{ tubes}$$

Therefore, make the condenser 2-pass

$$\text{Outlet water temperature} = 65 + \frac{235}{15.7} = 80\text{F}$$

$$\text{Mean temp. difference} = \frac{(104-65)-(104-80)}{\ln \frac{(104-65)}{(104-80)}} = 31.0\text{F (arithmetic average 31.5F)}$$

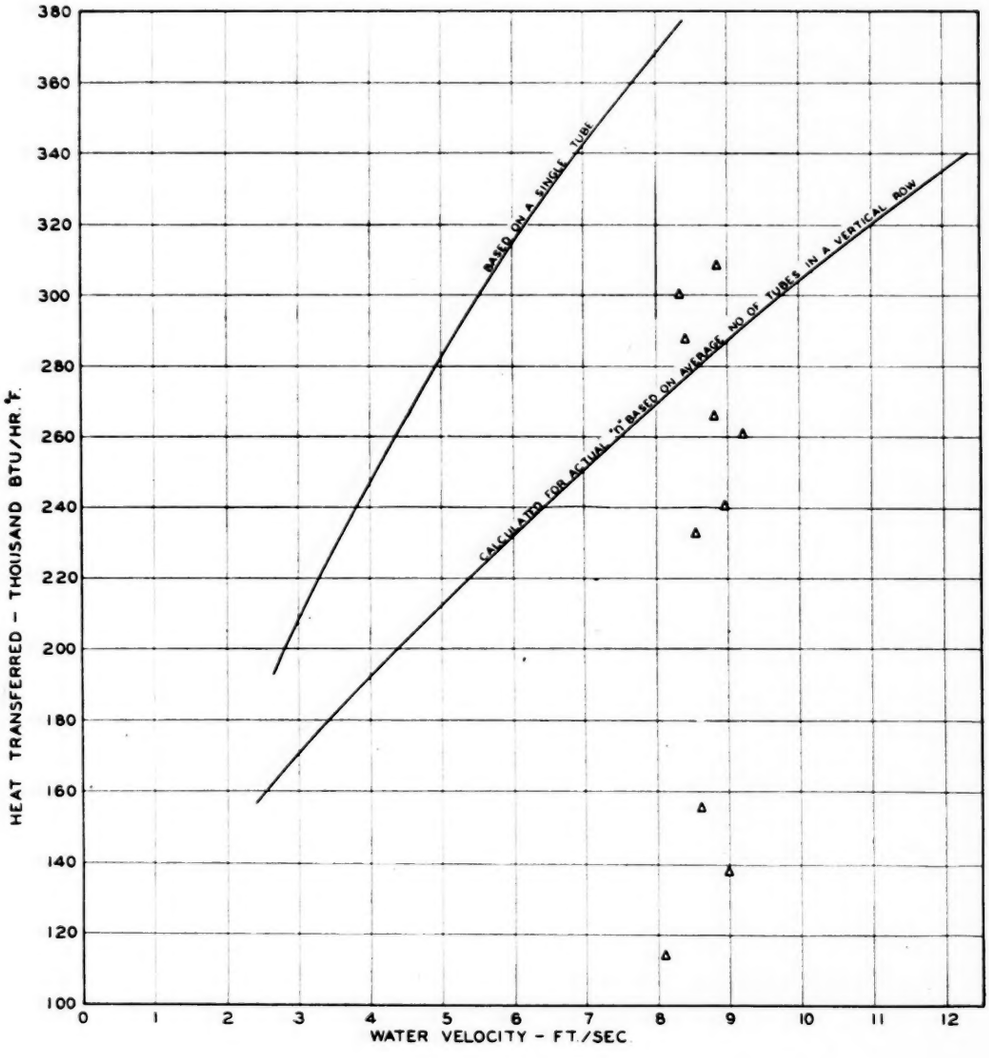

HEAT TRANSFERRED - THOUSAND BTU/HR. ft.
WATER VELOCITY - FT./SEC
BASED ON A SINGLE TUBE
CALCULATED FOR ACTUAL "G" BASED ON AVERAGE NO OF TUBES IN A VERTICAL ROW

Fig. 9. Experimental points and calculated curves for condensing Freon 11 in a condenser having 1168 sq ft area.

Now $q = U_o A_o \Delta T = 846000$ Btu per hr

Therefore $A_o = \frac{846000}{31.0(211)} = 129$ sq ft

Then total length of tubing $= \frac{129}{0.508} = 254$ ft

Length of condenser $= \frac{254}{2(26)} = 4.88$ ft

Method 2. Using Film Coefficients Assuming Clean Tubes

Assume the condenser has 52 tubes. Find the new area required.

Average water temperature $= \frac{65 + 80}{2} = 72.5$F

Then from Fig. 4, again assuming a water velocity of 6 ft per sec, the water-side coefficient = 365 Btu per (hr) (sq ft) (F). The resistance of the metal may be neglected. Since the condensing film coefficient will be under 600 Btu per (hr) (sq ft) (F) the thermal conductivity line of Fig. 7 indicates that the fin efficiency may be assumed to be 100% with only 2% error, and the actual outside area can be used.

The condenser has 52 tubes placed on a triangular pitch. From Equation (10):

$$n_{avg} = 0.40\sqrt{52} = 3 \text{ (approx.)}$$

$$n_{avg}^{0.25} = 1.32$$

Calculate the outside film coefficient using Fig. 3. Overall mean temperature difference = 31.5F (calculated in Method 1). Assume $\Delta T_{vf} = 16$F; then from Fig. 3, $h_o = 495$ Btu per (hr) (sq ft) (F)

Then from Equation (12):

$$h'_o = \frac{495}{1.32} = 375 \text{ Btu per (hr) (sq ft) (F)}$$

and from Equation (3)

$$\frac{1}{U_o} = \frac{1}{365} + \frac{1}{375} = 0.00274 + 0.00267 = 0.00541$$

But the temperature drop should be proportional to the resistance; i.e.,

$$\Delta T_{vf} = \frac{0.00267(31.5)}{0.00541} = 15.6\text{F}$$

This does not quite check, so assume $\Delta T_{vf} = 15.5$, then from Fig. 3, $h_o = 498$ Btu per (hr) (sq ft) (F)

$$h_o' = \frac{498}{1.32} = 377 \text{ Btu per (hr) (sq ft) (F)}$$

$$\text{and } \frac{1}{U_o} = \frac{1}{365} + \frac{1}{377} = 0.00539$$

$$\Delta T_{vf} = \frac{0.00265(31.5)}{0.00539} = 15.5\text{F}$$

$$\text{Then } U_o = \frac{1}{0.00539} = 186 \text{ Btu per (hr) (sq ft) (F)}$$

and from $q = U_o A_o \Delta T$

$$A_o = \frac{846000}{186(31.0)} = 147 \text{ sq ft}$$

$$\text{Total length of tubing} = \frac{147}{0.508} = 289 \text{ ft}$$

$$\text{Length of condenser} = \frac{289}{2(26)} = 5.55 \text{ ft}$$

Note: Use of the actual average number of tubes in a vertical row and its correction of the condensing film coefficient from 498 to 377 in the above calculation is a very conservative procedure, thus the condenser has an area some 13% greater than would be predicted by the experimental coefficient.

Method 3. Using Film Coefficients Assuming Fouling Inside the Tubes

From the T.E.M.A. Tables, resistance for Great Lakes water = 0.001 (hr) (sq ft) (F) per Btu.

Based on the outside area of the finned tubes in this condenser, this becomes

$$h_s = \frac{1000}{3.56} = 281 \text{ Btu per (hr) (sq ft) (F)}$$

Assume $\Delta T_{vf} = 7.5\text{F}$, then from Fig. 3

$h_o = 592$ Btu per (hr) (sq ft) (F)

Then from Equation 12,

$$h'_o = \frac{592}{1.32} = 449 \text{ Btu per (hr) (sq ft) (F)}$$

$$\frac{1}{U_o} = \frac{1}{281} + \frac{1}{365} + \frac{1}{449} = 0.00853$$

$$\text{Then } \Delta T_{vf} = \frac{0.00223(31.5)}{0.00853} = 8.2\text{F}$$

This does not check.

Assume $\Delta T_{vf} = 8.5\text{F}$, then from Fig. 3

$$h'_o = \frac{575}{1.32} = 436 \text{ Btu per (hr) (sq ft) (F)}$$

$h_o = 575$ Btu per (hr) (sq ft) (F)

$$\frac{1}{U_o} = 0.00356 + 0.00274 + 0.00230$$

Nomenclature

- A = Area in sq ft; A_o—outside area; A_i—inside area; A_m—average tube wall area; A_l—outside area per foot of tube; A_f—area of fins; A_t—horizontal area of a finned tube.
- D = root diameter of tubes in feet.
- g = gravitational constant.
- h = heat transfer coefficient in Btu per (hr) (sq ft) (F); h_o—based on total outside area; h_v—calculated for vertical surfaces; h_m—calculated for horizontal surfaces; h_w—water side coefficient; h_s—scaling coefficient.
- k = thermal conductivity in Btu (ft) per (hr) (sq ft) (F)
- l = tube wall thickness in feet.
- L = average height of a fin in feet.
- n = number of tubes in a vertical row in a multi-tube condenser.
- N = total number of tubes in a condenser.
- q = heat transferred in Btu per hr.
- U = overall heat transfer coefficient in Btu per (hr) (sq ft) (F); U_l based on area per foot of tube; U_o based on a square foot of outside area.
- ϕ = fin efficiency.
- ρ = density in lb per cu ft.
- μ = viscosity in lb per (ft) (sec)
- H = latent heat of vaporization in Btu per lb.
- ΔT = temperature difference in F; ΔT_{vf}—across the vapor film; ΔT_{oa}—overall temperature drop.

$$\frac{1}{U_o} = 0.00860$$

$$\text{Then } \Delta T_{vf} = \frac{0.00230(31.5)}{0.00853} = 8.5\text{F}$$

$$U_o = \frac{1}{0.00860} = 116 \text{ Btu per (hr) (sq ft) (F)}$$

$$\text{Area required} = \frac{846000}{116(31.0)} = 235 \text{ sq ft outside area}$$

$$\text{Total length of tubing} = \frac{235}{0.508} = 462 \text{ ft}$$

$$\text{Length of condenser} = \frac{462}{2(26)} = 8.89 \text{ ft}$$

A similar calculation for plain tubes, using the same fouling factor, indicated that about 62% more footage of tube is required. Assuming that tubes are 40% of the cost of the plain tube condenser, and that finned tubes cost 30% more per linear foot than plain tubes, the plain tube condenser would cost about 45% more than the finned tube condenser.

Experience is my Classroom

By T. W. REYNOLDS

HEAT PUMPS

The writer has lived long enough to know that 90% of our predictions and worries about things to come never come true. When current events upset our established customs in engineering it is natural, though, for someone to get upset and make dire predictions, but they don't know what they are talking about anymore than the writer does when he says they don't.

The war ended so much differently than foretold. As for the predicted boom, it is still just around the corner; indeed a mild recession is forecast. Who could forsee the effects of politics and the labor unions. So with the introduction of the heat pump and radiant heating; we can still look ahead without too much of fear as to the future of the radiator and other dear old things.

It was many many years ago that the heat pump and radiant heating were predicted as the things of tomorrow, but they did not come with the morrow, not because the ideas were devoid of merit, but rather because there weren't sufficient interests back of them for an expensive investigation and development, nor was there a rational application. In any event, they are finally here—and how. Whether they will join hands with the boiler and radiator and live happily ever after, time alone will tell. Perhaps these newer things will be supplanted by even newer things before they have even supplanted the old ones.

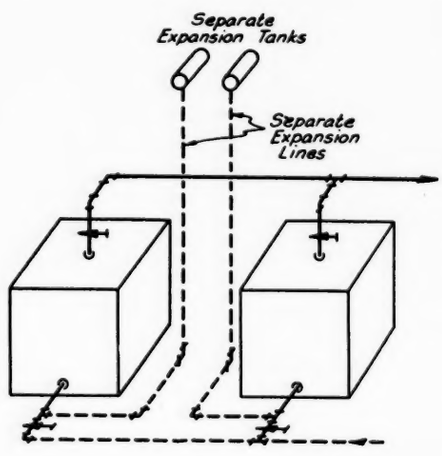
Where will all this lead us to? One can imagine many things. For example, they long investigated wave motor driving heat pumps along our sea coast thus forcing the utilities to other fields inland for an all-year electric load with the heat pump. Or would an atom bomb at the polar cap help them out by reversing the Gulf Stream, so that more heat from the air or ground could be obtained in the northern states. And there are hopes of higher ground temperatures for the heat pump to draw upon, temperatures which, according to one scientist, have been rising since 1850 and at an accelerated rate since 1920.

Back in 1917 the writer measured ground temperatures in unheated wide storage structures in Canada, and found that the ground was heating them to temperatures above freezing, temperatures which dropped only slightly by the close of the winter season. Why not heat a well-insulated house direct from borings in the ground without a heat pump at all?

EXPANSION CONNECTIONS

A spare boiler in a greenhouse hot water heating system is a wise precaution, providing the boilers are properly connected for the valving off of either one, and there is no interference with expansion of the water from either boiler. As illustrated, there must be a separate expansion line for each boiler outside of its shut-off valves. Each boiler should also have its own expansion tank, since two expansion lines connected to a common tank might siphon the water from the active boiler over to the idle boiler, if and when the idle boiler is being drained of water.

Obviously, although the boilers should be valved off from one another so that one can be laid up, the valves are also required for another purpose. Without them there would be a heat loss from the idle boiler as the hot water from the active boiler would be circulated backwards through the idle boiler. This backward flow would also oppose the return circulation to the active boiler.



Expansion connection, for twin boilers

RADIATOR PRESSURES

Because of limitations in the manufacturer's guarantee on cast iron radiation, many users of such have the impression that cast iron is an inherently dangerous material for the confining of steam or water under pressure. The guarantees (as are well known) are for 15 lb steam and 30 lb hot water working pressure, but working or operating pressures are not always the actual pressures within a system. For example, condensate from steam can get trapped near a fitting where the resulting water hammer will do more multiplication in pressure than a rabbit could in progeny.

Manufacturers of cast iron radiation will guarantee their radiators against breakage or leakage for much higher pressures than the usual, providing the purchaser will guarantee in turn that the actual pressure will never exceed the higher pressure for any reason whatsoever. The manufacturer upon request will also test the radiators up to 400 lb and give a certificate of pressure to that effect. Radiators are commonly tested at about 100 lb pressure and when tested to destruction only begin to break around 800 lb.

Castings are, of course, subject in some cases to unequal cooling or a thin spot due to a shift in the core. Such a defect might not show up on test, but could develop a break or leakage due to piping strains in actual installation. In manufacture, however, there are only about 3% which are found defective after casting (no breakage) and practically all of this occurs in the first or section test and not in the assembly test. Limitations in manufacture as to thinness of metal and cores is the reason why cast iron radiators take up so much space.

Users of cast iron radiators are sometimes puzzled by the differention in pressure guarantees, i.e., steam at 15 lb, water at 30 lb. The reason is simple, however, since any steam boiler operated at over 15 lb pressure requires a licensed engineer. A second reason for differention could be the action of water hammer as before mentioned.

The effect of damage to life by breakage of radiators is entirely a matter of heat release. The hot water of a forced circulation might be at a temperature of 250F and therefore could flash quite a bit of heat down to 212F. Hot water can also do a lot of water damage.

Note: Mr. Reynolds is Chief, Air Conditioning Division, Abbott, Merkt & Co., Engineers, New York.

Radiant Heat with Copper Tubing —

Fabrication of Heating Coils

D. L. MILLS and L. J. LA TART
Research Department, Revere Copper and Brass, Incorporated

This is the second of a series of articles covering the radiant heating installation at the metallurgical laboratory of Revere Copper and Brass Incorporated. The system, of which a general description appeared in last month's issue, will be used as a source of fabrication, performance, and heat transfer data, as well as serving the laboratory building as a heating system. This month, fabrication techniques are described.

As a result of the increasing number of radiant heating installations, a larger number of heating contractors are seeking practical information regarding many of the installation details that are commonly necessary. It is with this in mind that this article is written.

WHEN the contractor is planning his installation procedure, early consideration, undoubtedly, will be given to methods of fabricating the coils that are to be embedded. If the contractor has a choice of materials to be used for the piping system, he will naturally make his selection on the basis of high thermal conductivity of the metal from which the piping is made (bearing in mind that when coils are embedded no radiation or convection from the surface of the piping takes place) its workability, ease of handling, weight, lengths available, and cost of making the necessary joints and connections.

Copper tube combines so many advantageous features as a piping material for radiant heating coils that methods of capitalizing on all these advantages should be of interest to many contractors.

In the first place, it will generally be desirable to purchase copper tube in soft temper, 60-ft coils. This length is commonly carried in stock by distributors and reduces the number of couplings required for the ordinary sinuous coil. In planning the size of the coils as to length of run, the pressure drop through each coil should be determined and held to a proper amount. This will depend in general on the flow rate through the coil, the length of the mains, the size of the tube used for the coils and the pump capacity.

Field fabrication of sinuous coils for radiant heating is a comparatively simple matter if copper tube is used. In the building described in a previous article (H&V: 10:47:p 65), a simple bench (Fig. 1) was constructed 36-in. wide by approximately 25 ft long. On top of the bench, narrow wood strips of ½-in. material were nailed parallel to the long edge of the bench. These strips were spaced the proper distance apart to accommodate the copper tube so that it would be straight as it was unrolled from the circular coil in which form it was shipped. The center to center distance between

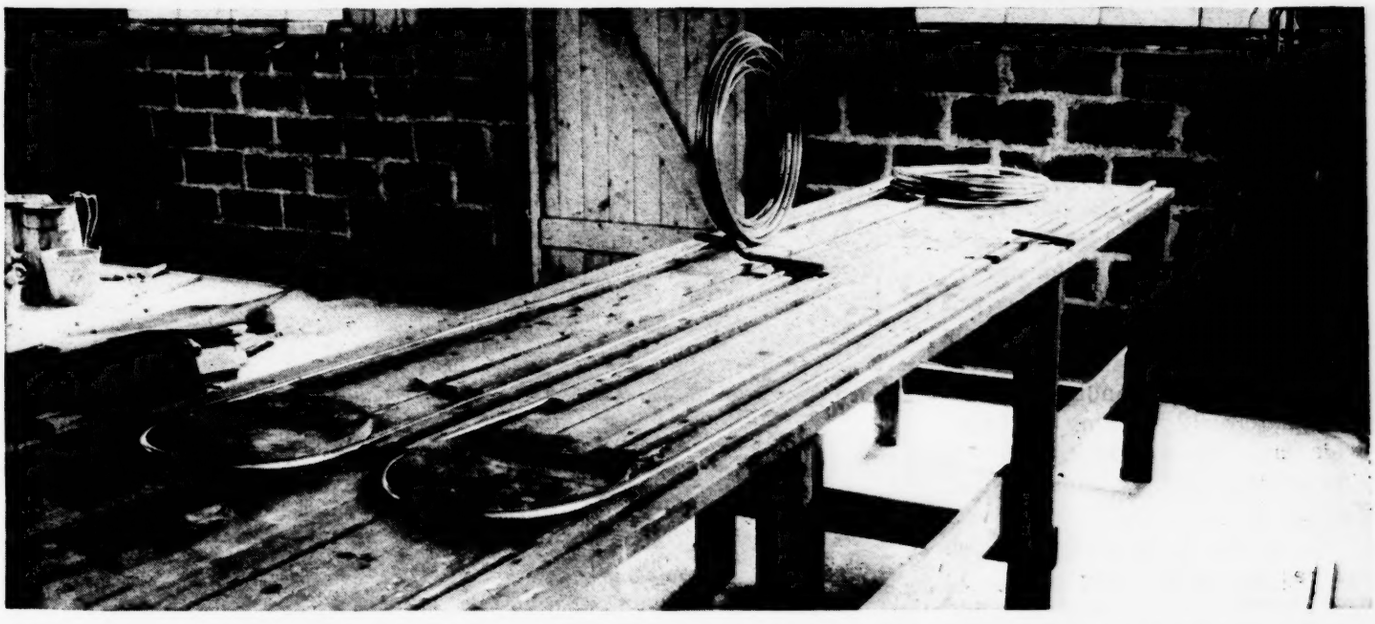


Fig. 1. Temporary bench with coil partly formed. Bench is made wide enough to accommodate two large return bends.

Fig. 2. Metal sheaves and handle for bending.

each pair of strips corresponded to the center to center spacing of the tubes in the heating panel.

Wood discs made of ¾-in. material were cut on a band saw with a radius equal to the radius of the inside of the return bends of the finished coil. These discs were used in forming return bends and were fastened to the bench so as to line up with the spacing strips. They were spaced at a lengthwise distance to correspond with the length of the finished coil.

In this particular installation, large-radius bends were altered with return bends on 2-in. radii. To facilitate the bending of the copper tube on a 2-in. radius, a metal sheave approximately ¾-in. thick was made and was fastened to the top of the bench. (Fig. 2). Around the perimeter of the sheave, a groove was cut corresponding to one-half the outside diameter of the copper tube. In the center was a hole to accommodate a fixed pin on the handle used in the bending operation. This handle was approximately 12-in. long by 1-in. wide and ⅜-in. thick, as this seemed to be heavy enough stock for use in this case. At one end of the handle was a fixed pin which entered a centrally located hole in the sheave referred to above and acted as an anchoring center.

A second sheave similar to the first but approximately 2 in. in diameter was made. This sheave was held in position by a second fixed pin in the handle of the bending device which entered a centrally located hole in the small sheave. The tube was bent by drawing the handle radially so that the small roller sheave would revolve along the tube and keep it in contact with the groove in the fixed sheave as it was being bent around the larger sheave. These sheaves are necessary for bending Type L copper tube on short radii so as to prevent the tube from collapsing.

When the bend is about to be made the bending wheel is brought up to position in contact with the tube and the handle is then put in place by having the two fixed pins engage the hole in the fixed sheave and in the small sheave. The latter will in general be considerably smaller in diameter than the fixed sheave.

In starting to form the sinuous coils the copper tube was unrolled on the bench from the 60-ft coil in which form it was furnished. The unrolling was done between the guide strips on top of the bench, thus keeping the tube straight. Opposite the starting end of the bench the copper tube was uncoiled around the wood disc (Fig. 3) and held tightly against it. As the tube was bent around the disc it entered the second pair of guide strips and the uncoiling of the tube continued until the starting end of the coil was reached. At this point the larger of the metal sheaves was located (Fig. 4) to form the short radius return bend. The bending operation was then repeated and more tube was unrolled until the complete heating coil was formed, ready to be put into position.

At each end of the bench as many wood discs were located as the width of the bench would accommodate (Fig. 1) depending on the tube spacing which, of course, determined the size of the discs. When the

Fig. 3. Forming large return bends of coils.

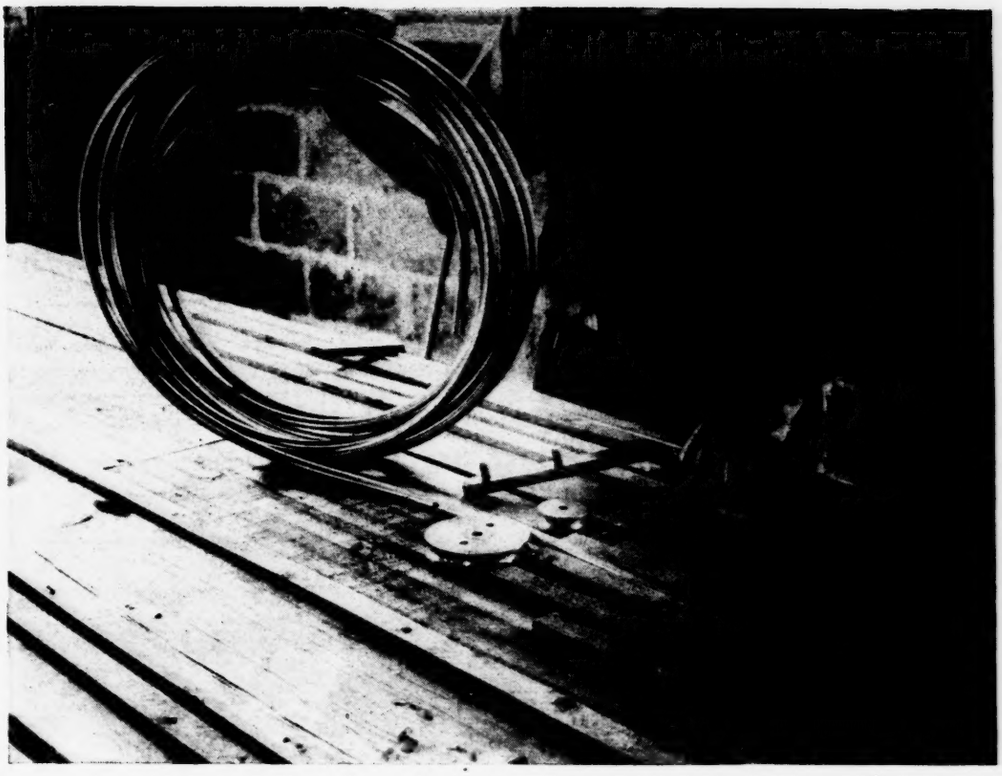


Fig. 4. Small return bend about to be formed.

end of the 60-ft coil was reached, a solder type coupling was applied right on the bench and the uncoiling of a second coil of copper tube continued. In cases where the sinuous coils were considerably wider than the bench, extension pieces were applied to accommodate the coils temporarily.

As a means of easily handling the completely formed coils, a light wood frame (Fig. 5) was constructed of 1-in. by 2-in. material. On the cross members of this frame were small blocks of wood which held the copper tube in proper alignment and spacing. After the bending had been completed, the coil was raised slightly and the frame was slipped under the coil on top of the bench. The copper tube was then wired to it temporarily in four places (Fig. 6). It was then an easy matter to handle the coil. In the case of the ceiling coils, two men could carry the frame with the coil on it, raise it to the scaffolding platform and temporarily wire it into position on the ceiling. As soon as this was done the copper tube was permanently wired in place. After the copper tube had been wired to its supporting members in a sufficient number of places the wood frame was taken down and the wiring operation was continued until completed.

It was found that the use of the wood frame greatly facilitated the handling of the coils after they were formed. It not only saved time but it eliminated the necessity of measuring the spacing of the rows of tubes, as the tubes were all properly spaced when they were placed in position.

Furthermore, forming the coils on the bench was found to be much easier and quicker than attempting to unroll the copper tube on the ceiling and bend it overhead.

A method of fabrication similar to that described above was used for constructing the floor coils. It was found, however, that in bending the ¾-in. copper tube a little work-hardening took place so that at the completion of the bending, the coils were not as flat as could be desired. In order to flatten the coils, each completed coil was placed on a portion of the concrete floor that had been poured and was suitable for traffic. In such cases the frame was on top of the coil as it lay on the floor. This was the reverse of the position of frame and coil when ceiling coils were being handled. In this way, blocks of wood could be used in hammering

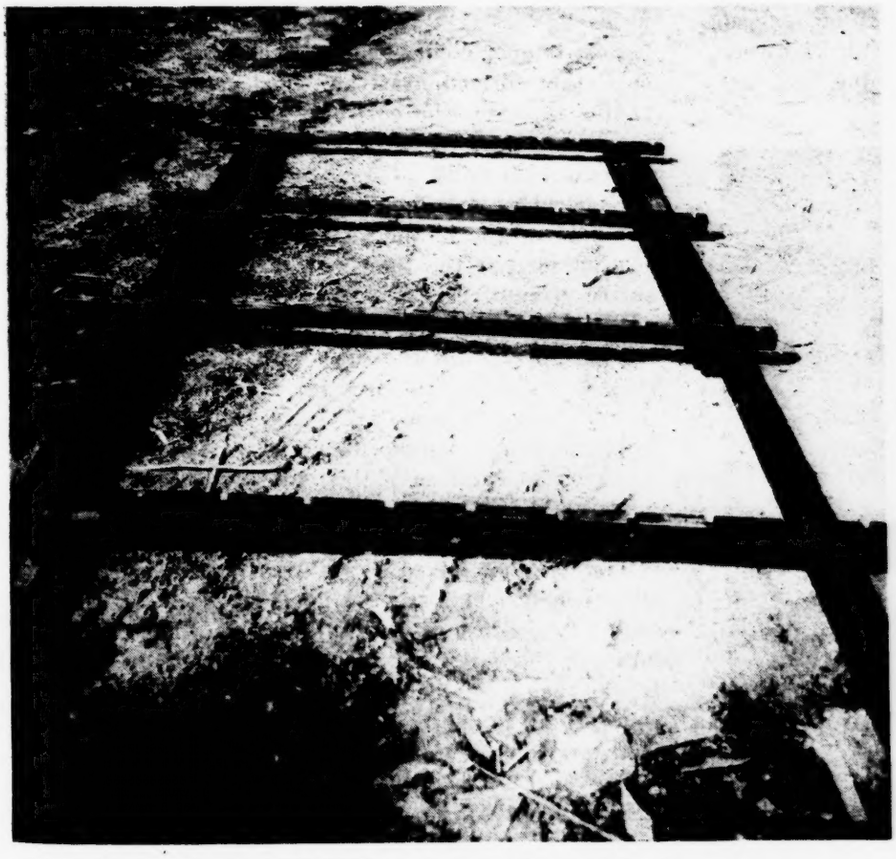


Fig. 5. Wood frame for ⅜-in. copper tube spaced 4 in. on centers.

Fig. 6. ¾-in. copper tube on 9-in. centers temporarily wired to wood frame.

coils flat without damaging the copper tube in any way. When the coil was flattened it was transported to its proper location on the floor. When the coil was unwired from the frame, the copper tubes rested on insulated wire saddles (Fig. 7) that held them at the proper depth below the finished floor surface.

These saddles were devised for this installation as a means of assuring extreme accuracy in depth of bury for test purposes. Undoubtedly, other means of properly supporting coils in a floor slab can be devised, but in any case insulation should be provided between the coils and the supports to prevent undue heat loss through the supports if they are of material that has high thermal conductance value.

Fig. 7. Wire saddles support copper tube at desired distance below finished floor surface.

It should be noted that the wood discs referred to above, used for forming the return bends of the coil, will be limited as to radius by the size of the copper tube to be bent and by the center to center spacing. If the radius of the return bend is short and there is danger of the copper tube collapsing in the bending operation, metal sheaves already described should be used.

Soft temper copper tube can be bent by hand in its cold state on comparatively short radii, but when the radius is short, the entire circumference of the tube must be supported by the bending sheaves in order to prevent collapsing of the tube. In such cases the peripheral groove in each sheave must coincide with one-half of the outside surface of the tube in cross section so that the tube wall is supported around its entire circumference.

When forming the coils, a sufficient length of the straight portion of the tube should be left for connections to mains or branches. It has been found advisable in most cases to place all of the coils in their proper locations before the mains and branches are installed. By doing so, sections of the mains and branches can be made right on the bench by soldering the tees, elbows, or other fittings in their proper places.

Time can be saved by fabricating branch headers on a bench. This can be done in most cases where solder type fittings are used. All such header assemblies for the building described were bench fabricated. It was then necessary to make only the few remaining connections to the coils and ends of headers in their final location.

The next article in this series will deal with the subject of tube location and its relation to the panel surface with particular reference to plastering and methods of support.

Methods of Joining Pipe –

Welded and Brazed Pipe Joints

J. E. YORK

Building Service Engineer, Stone & Webster Engineering Corporation

The advantages and limitations of welded joints as methods of joining pipe, and the use of brazed joints are discussed in this article. The common forms of welds and fittings are illustrated and described in detail.

USE of welded joints has steadily increased during the last 15 or 20 years and during the last 5 years this type of joint has been used extensively.

Research and practical experience in the shop and fields have developed the technique of welding to such an extent that when done by qualified welders under proper procedure control, its use is no longer questioned.

Types of Welded Joints

There are numerous types of welded joints of which those in general use are:

Butt welds for connecting together beveled ends of pipe—Fig. 1, 2 and 3

Butt welds using welding rings—Fig. 5

Cup welds—Fig. 6

Sleeve welds—Fig. 7

Welding nozzles—90° and angular standard type—Fig. 8

Welding nozzles—90° and angular reinforced type—Fig. 9

Mitre welds for fabricating bends

Welded end closures—Fig. 14, 15 and 16

Welding fittings—Fig. 4

Advantages of Welded Joints

Welding offers the following advantages for joints in a piping system:

Greatest composite strength because all parts are permanently built into the systems.

Light weight with resultant simplification of supporting structure and method of suspension.

Closer spacing between pipes and between pipes and structures.

Neater appearance.

Simplification of the application of insulation.

Lower maintenance costs and less time out of service for repairs to leaky joints.

Ease with which branch connections can be added to existing headers or piping.

Adaptability for use with multiple outlets, drip pockets and special angle connections.

Lower cost than for flanged joints on high pressure lines.

Limitations of Welded Joints

Welded joints for piping sytems are not advisable for the following conditions:

Where valves, strainers, regulators or other devices are used which must be removed occasionally for repairs or replacement.

In locations where the use of a welding torch might cause a serious hazard of fire or explosion.

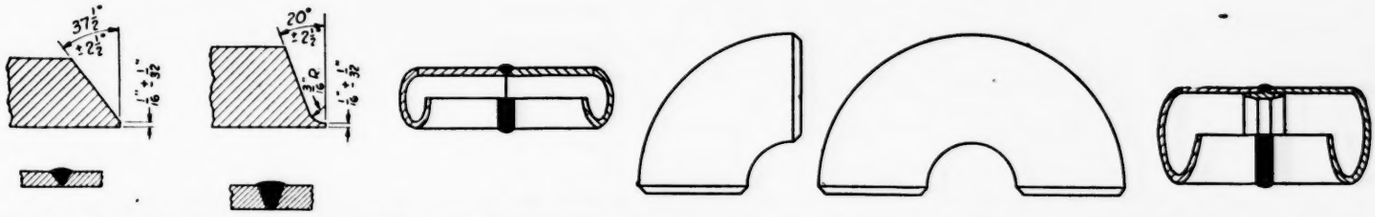
In localities where it may be economical to use this type of joint for initial installation, but skilled welders and welding equipment are not available for emergencies.

Butt Welding

The ASA standard for butt welding (ASA B16.3) requires that for butt welding of two sections of pipe, or pipe to welding neck flanges, welding fittings, welding end valves, etc., with metal thickness of ¾ in. or less, a straight bevel of 37½° shall be provided with a straight face of 1/16 in. as shown in Fig. 1.

When the thickness is greater than ¾ in., the U-bevel, Fig. 2, shall be used for electric welding only. For acetylene welding the plain 37½° bevel shall be used regardless of thickness.

Bevels at an angle of 45° are normally considered as special and are only furnished when so ordered and at extra cost.



Left to right, Fig. 1. Bevel and butt weld for metal thickness ¾ in. or less. For greater metal thickness see Fig. 2. Fig. 3. V-type butt weld. Fig. 4. Two types of standard weld fittings. Fig. 5. V-type butt weld with welding ring.

Fig. 6 (left) Section of a cup weld.
Fig. 7. Detail of sleeve weld.

After the ends of two pieces of pipe have been correctly beveled they should be thoroughly cleaned, and then brought together until they nearly touch. After the alignment has been carefully checked, larger sizes of pipe should be tack welded before making the complete weld but this tacking may be omitted on the smaller sizes. See Fig. 3. Two of a complete line of welding fittings are shown in Fig. 4.

When making a welded joint in this manner there is a very great possibility that icicles or welding pellets will get into the line. Where the inside surfaces can be reached after the welding operation it is good practice to grind these surfaces to remove all icicles, and where made in a closed section of pipe it is necessary to thoroughly blow out the system before attaching valves or other equipment which might be damaged by particles of welding material when the system is placed in operation.

To eliminate the possibility of icicles or welding pellets getting into the line, butt welds are often made with welding rings inserted inside the pipe at the point where the weld is to be made as shown in Fig. 5. They also insure correct spacing and alignment of the pipes, or pipes and fittings which are to be welded together.

To eliminate any restriction in pipe area by use of welding rings as shown in Fig. 5 the ends of the pipe are sometimes expanded and a sleeve which has approximately the same internal diameter as the pipe is used.

Cup welding (Seigle joint, Fig. 6) is sometimes used for field welding as this type of joint is self-centering; eliminates the use of clamps for holding the pipe in place as is customary with regular butt welds; and prevents the formation of icicles on the pipe interior.

Sleeve welds, Fig. 7, are recommended where unusual stresses are encountered. There are many variations of this type of joint. Some engineers require, in addition to the two welds shown in the illustration, that the reinforcing sleeve be split and longitudinally welded. Others require spot welds around the center line of the sleeve.

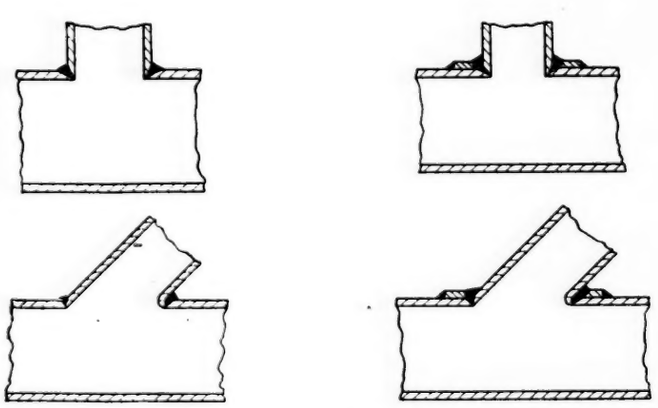


Fig. 8 (left) 90° nozzle weld and angle nozzle weld, both regular type. Fig. 9. 90° nozzle weld and angle nozzle weld, both reinforced type.

Socket type welding fittings, Fig. 10, prevent the possibility of formation of icicles and insure a strong sound weld, with assurance of correct alignment of pipe sections.

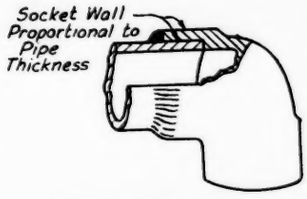


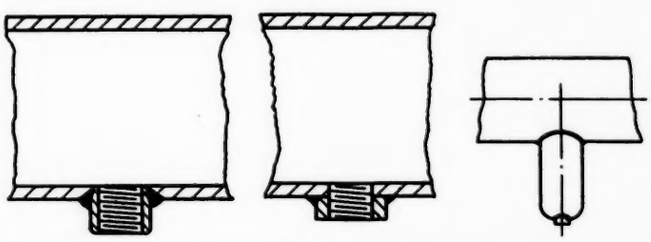
Fig. 10. Socket type.

A 90° nozzle weld, regular type, and an angle nozzle weld, regular type, are shown in Fig. 8. Fig. 9 shows 90° nozzle weld, reinforced type, and angle nozzle weld, reinforced type.

For all of these welds, both main line and outlet, connections are beveled and should be thoroughly cleaned prior to welding. Nozzle welding is a continuous process and a joint once started should be completed as quickly as possible. A weld should not be permitted to cool in a partially completed stage. For normal requirements the regular type nozzle is satisfactory.

Fig. 11 shows a coupling for screwed connection welded into the line, and Fig. 12 a pad welded to the pipe, and drilled and tapped for screwed connection.

Fig. 13 illustrates a drip pocket welded into a steam line. Fig. 14 and 15 show flat head and dished head



Left to right. Fig. 11. Screwed connection welded to line. Fig. 12. Pad welded to pipe. Fig. 13. Drip pocket welded to steam line.

closures which are made by welding a piece of plate inside the end of the pipe. A welding cap, Fig. 16, is preferable to either of the arrangements pictured in Fig. 14 and 15.

Fig. 17 shows the relative stresses in a pipe and a welded elbow of uniform thickness. To offset the high stresses developed on the crotch of the elbow, it is the practice of manufacturers of high grade welding fittings to thicken up the metal at this point, as in Fig. 18.

It is also desirable to have a tangent piece on each end of an elbow to remove the weld from the curved section.

It is sometimes necessary to use mitered welds in place of fittings, but when properly designed fittings can be obtained, their use is generally preferable. The type fitting in Fig. 19 is available for use on pipe mains from 1 in. to 12 in. dia with branch outlets from ½ in. to 6 in. dia. Their use eliminates the necessity of making preliminary layouts of the main line and branch, cutting holes to template, and preparing the connecting branch by cutting and beveling with a torch. They are installed by welding to the main, using a V-weld, then removing the button and completing the junction by welding the branch pipe to the fitting.

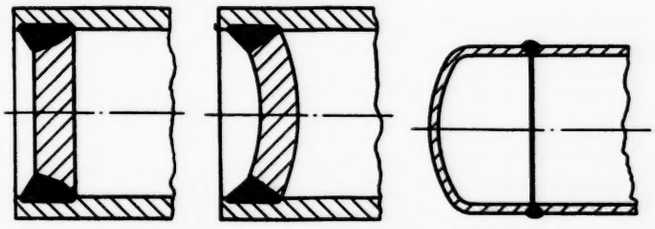
A similar type of fitting is used with screwed pipe for the branch connection. Care must be exercised when using this type of fitting to prevent distortion of the pipe thread. This makes it impossible to provide a tight screwed connection.

Brazed Joints

The process of making joints between copper, brass or bronze pipes and fittings similar to welded joints between steel and wrought iron pipe is called brazing. Brazing is done with an oxyacetylene flame as the source of heat. Brazed joints if properly made should have a tensile strength, equal to approximately three times the strength of standard weight brass pipe.

They are used satisfactorily on hot and cold water service lines, vent and drainage lines, boiler feed lines for 400 lb maximum pressure, process water, steam, air and chemical lines and for marine piping for 150 lb and 250 lb pressures. They are also used for refrigeration lines, and by the railroads for fuel oil, lubricating oil, etc.

When making brazed joints between adjacent lengths of piping, the ends should be beveled as described for welding steel or wrought iron pipe.



Left to right, Fig. 14, 15 and 16. Flat head, dished head and welding cap form of welded end closure.

When brazing copper pipe a copper brazing rod should not be used. Bronze brazing rods should be used with either copper or brass pipe. Brazing alloys having high melting temperatures should be used. A silver brazing alloy furnished by one manufacturer has a flow point of 1300F.

One of the leading manufacturers has developed a line of fittings, valves and flanges for making silver brazed joints. The ends of the fittings, valves and flanges are bored to provide a very small clearance area between the fitting and the pipe, and this space is entirely filled with silver brazing alloy upon completion of the joint. The fittings as supplied by the manufacturer have a ring of braze alloy in a groove provided for the purpose near the edge of the fitting. The silver brazing alloy has a high melting temperature and its corrosion-resisting qualities are the same as the parts joined.

These joints are designed to withstand tension, compression, vibration and corrosion to the same degree as the pipe or tube with which they are used.

The following procedure must be carefully followed when making this type of joint:

The pipe used must be of nonferrous material having an outside diameter corresponding to that of standard size pipe and must not be out of round. Pipe must be cut square and must be free from burrs or upsets.

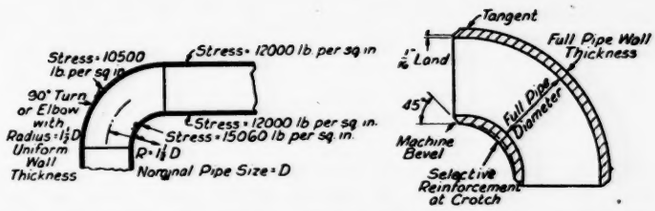

Stress = 10500 lb per sq in
Stress = 12000 lb per sq in
90° Turn or Elbow with Radius = 1½ D Uniform Wall Thickness
Stress = 12000 lb per sq in.
Stress = 15060 lb per sq in.
R = 1½ D
Nominal Pipe Size = D
Tangent
Full Pipe Wall Thickness
¼" Land
45°
Full Pipe Diameter
Machine Bevel
Selective Reinforcement at Crotch

Fig. 17 (left) Stress in elbow and pipe when internal pressure develops 12,000 lb per sq in. stress in straight pipe wall. Fig. 18. How crotch is reinforced to offset high stresses.

The end of the pipe must be cleaned with emery cloth for a distance equal to the length of the fitting recess.

The fitting outlet must be thoroughly cleaned.

The end of the pipe or tube and the inside of the fitting recess must be thoroughly coated with flux as recommended by the manufacturer.

Pipe or tube must be inserted into the fitting up to the shoulders as shown in Fig. 20.

The pipe or tube must first be heated which causes it to expand and the flux to flow. The clearance area closes and the braze alloy contacts the heated tube surface.

The fitting is then heated, directing the flame toward the tube or pipe. This causes the fitting to expand, opening the clearance area and allowing the flux and alloy to flow.

The pipe or tube and fitting are then both heated, keeping the torch pointed toward the pipe or tube using a wiping motion for 15 to 20 seconds. The torch is then removed allowing the fitting to cool and contract, thus forcing the alloy to penetrate the surfaces and to form a fillet at the fitting edge.

The heating method outlined above requires the use of oxyacetylene flame. A quarter section only of the joint should be heated at a time. When the fillet is clearly visible apply heat to the next quarter section. The entire circumference of the joint should not be completed at one time. On pipe sizes ½ in. and smaller,

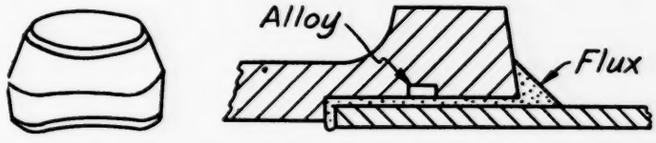


Fig. 19 (left) Fitting for use on pipe mains. Fig. 20. Fitting for making silver brazed joint is provided with a braze alloy ring.

best results are obtained by making the joint quickly and with a minimum amount of heat, because the alloy will flow out over the entire circumference. Excess heat by enlarging the clearance, retards rather than hastens the flow of the alloy. If in a section being heated the fitting or pipe becomes bright in color, heat should be withdrawn.

Joints must be well supported during the process and they must not be strained while cooling.

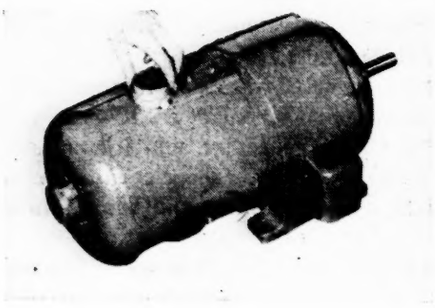
For ½-in. pipe the heating time required per joint is about 2/3 minute and from 40 to 50 joints can be made per hour.

For 6-in. pipe the heating time required per joint is about 7½ minutes and about 6 to 8 joints may be made per hour.

NEWS OF EQUIPMENT AND MATERIALS

Induction Motor

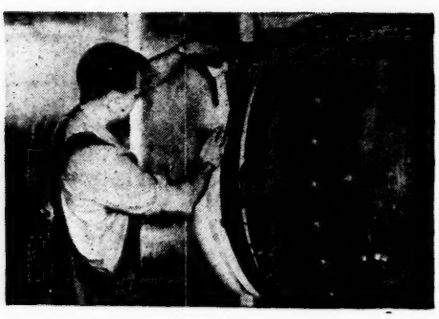
NAME—Tri-Clad adjustable speed induction motor.
PURPOSE—For general use where an adjustable speed motor is required.
FEATURES—Stepless speed adjustment is obtainable over the 3:1 ratio by turning a dial. The entire unit with the exception of the con-



trol is self contained. This type ACA motor is a constant torque continuous rated motor having shunt characteristics over a 3:1 speed range. Percentage drop in speed from no load to full load at maximum is 5 to 10%; at low speed, 15 to 30%. Remote control up to 10 ft can be accomplished by the use of a flexible cable shaft. For complete remote control, a small pilot motor is mounted on top of this unit to drive the speed control knob.
SIZES AND CAPACITIES—From 3 to 50 hp.
LITERATURE AVAILABLE—GEA-4854A.
MADE BY—*General Electric Co., Schenectady, N. Y.* **563**

Industrial Tape

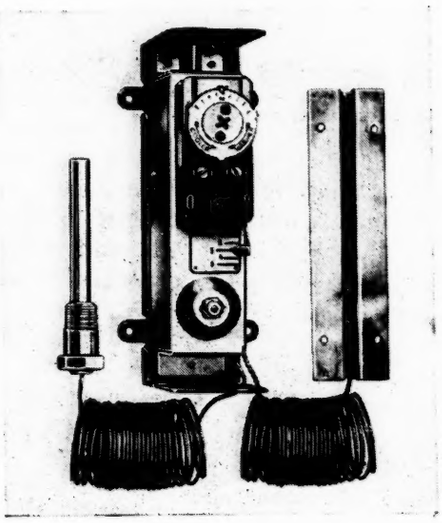
NAME—No. 263 Fiberglas industrial adhesive tape.



PURPOSE—For general needs where strong fabric tape is required for a firm, permanent seal.
FEATURES—Tape of Fiberglas cloth backing employs a vulcanizable adhesive. The thin backing is said to resist heat and will not shrink or stretch. Tape has been used as an air seal in the fabrication of flexible duct connectors.
MADE BY—*Bauer & Black, Division of The Kendall Co., 2500 So. Dearborn St., Chicago 16, Ill.* **564**

Temperature Control

NAME—White-Rodgers indoor-outdoor control.
PURPOSE—Temperature control for radiators, convectors, and radiant panel heating systems.
FEATURES—Control is operated by two sensitive bulbs, one of which is



installed outside the building to measure actual outdoor temperature, and the other, inserted into the heating system, measures the average water temperature. When weather is mild this control permits the heating system to operate at a lower water temperature; during periods of cold weather higher water temperatures are possible. Control has two adjustments; one to establish the water temperature as required for the comfort needs of the occupant, and the other adjusts the water temperature in accordance with outside weather conditions.
MADE BY—*White-Rodgers Electric Co., 1209 Cass Avenue, St. Louis, Mo.* **565**

Ring Packing

NAME—Smith-Blair water-lock ring.
PURPOSE—For packing a bell and

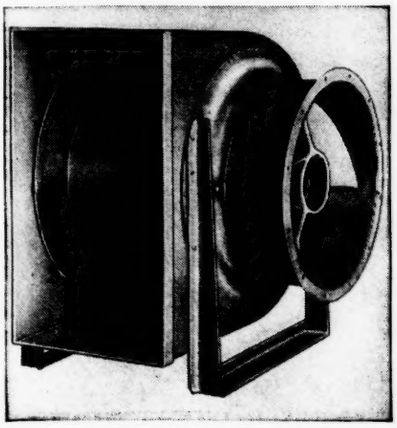
spigot cast iron pipe joint.
FEATURES—Made of molded rubber, it is installed by rolling the ring back over the spigot end of the pipe. It is designed to hold 40 to 60 lb pressure without a compound. Sealing action is increased from the pressure in the main.
MADE BY—*Smith-Blair, Inc., South San Francisco, Calif.* **566**

Humidity Control

NAME—Humidity sensing element.
PURPOSE—For determining the humidity of air.
FEATURES—Makers claim that element can retain an accuracy within the limits of plus or minus 1.5% relative humidity for at least one year. Element, which operates on the ability of a hygroscopic film to change its electrical resistance, is said to respond in less than one second to either decreasing or increasing relative humidities. Accuracy is not affected by changes in barometric pressure.
LITERATURE AVAILABLE—Bulletin 93-2140.
MADE BY—*American Instrument Co., Silver Spring, Md.* **567**

Axiflo Fans

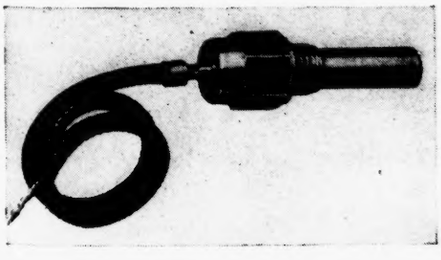
NAME—Westinghouse Axiflo fan.
PURPOSE — Air flow in connection with air conditioning, ventilation, dust removal, and drying processes.
FEATURES—Makers claim that this type fan has large volumetric capac-



ity to perform effectively against resistance of wind and duct systems. Unit has straight air flow and simplified mounting. The convertible elbow type with motor, belt and bearings located outside air stream, is readily accessible for maintenance and inspection.
SIZES AND CAPACITIES—From 18 to 72 in.; static pressures from 0 to 3 in.; displacement from 2000 to 115,000 cfm.
MADE BY—*Westinghouse Electric Corp., Sturtevant Div., Hyde Park, Mass.* **568**

Thermal Control

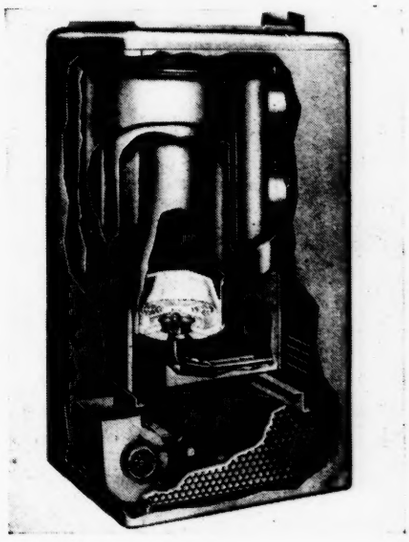
NAME—Plug-Stat.
PURPOSE—Thermal control and limit switch.
FEATURES—Originally developed as a thermal switch to warn when excessive temperatures were reached in automotive gear boxes, unit has adaptability to a wide variety of fields. Its thermal characteristics are control range, –60 to +600 F;



amplitude of operation, plus or minus 3F, electrical characteristics a-c or d-c, 28 volts, 2 amperes. Control was designed for severe vibration or dirt conditions. Switch weighs 0.7 oz.
MADE BY — *Control Products, Inc., 306 Sussex St., Harrison, N. J.*...**569**

Gas Burner

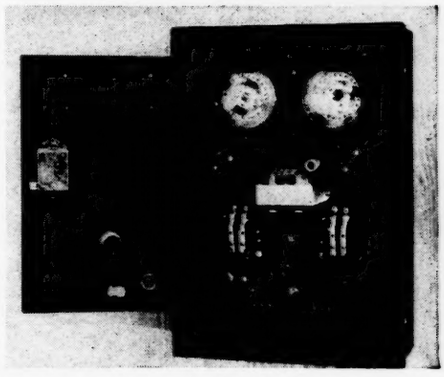
NAME—Iron Fireman gas conversion burner.
PURPOSE—Burner for converting heating systems to the burning of gas.
FEATURES—Burner has a heating element which becomes radiant to absorb the heat of the gas flame. Burner has horizontal gas ports in a stainless steel burner head. Head preheats air which is thoroughly mixed with the gas to give high



combustion efficiency and lower flue gas temperature.
MADE BY—*Iron Fireman Mfg. Co., Cleveland, Ohio.* **570**

Heating Control

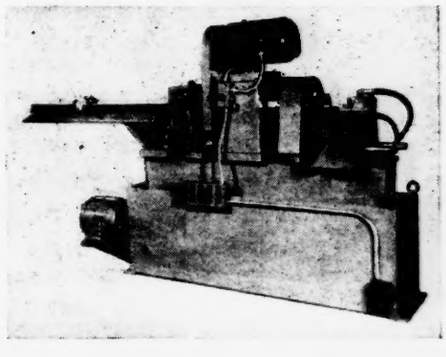
NAME—Fuel Watchman heating control.
PURPOSE—For automatic control of space heating.
FEATURES — Once installed and adjusted for individual building controls, unit is said to operate automatically to control daily temperatures without any additional adjustment. Device consists of a sun control, north side control, and main control panel. Sun control prevents overheating of the building on mild days. Outside control, placed on the shady side of the house, operates over a wide tem-



perature range. Main control panel is placed in the boiler room or at the heat source. On a hot water system this control regulates the circulating pump according to outside temperautre conditions.
MADE BY—*Fuel Watchman, 72-29 138th St., Flushing, L. I.* **571**

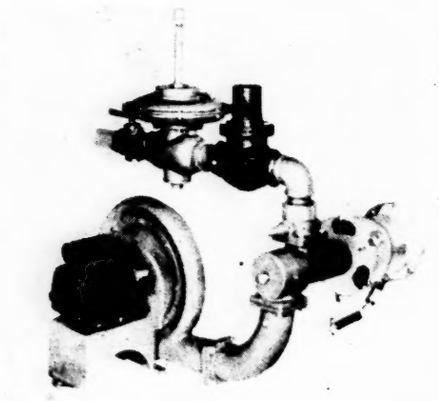
Pipe Cutter

NAME—Pines cut-off machine.
PURPOSE—Automatic machine for cutting pipe and tubing.
FEATURES—Work is fed by motor driven rolls through a hollow spindle and against adjustable receding target stop. A rotating head automatically cuts the work to required length. Head is tooled to produce a clean cut with a minimum of burs. Once the machine is started, operation is continuous until machine runs out of stock.
LITERATURE AVAILABLE — Catalog.
MADE BY—*Pines Engineering Co., Inc., Aurora, Ill.* **572**



Gas Assembly

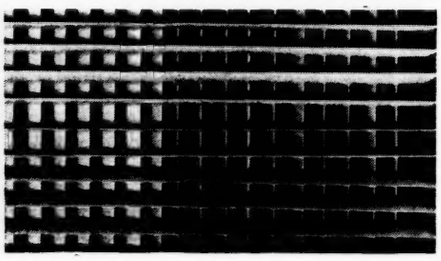
NAME—Bryant Pow-R-Semblies.
PURPOSE—Packaged units for use with boilers, dryers, and wherever semi-open burners with controlled flames are desired.
FEATURES—Unit has a newly developed proportional mixing device. Air is used from a constant-pressure blower and the device draws gas through a governor in whatever quantity is required for the mixture selected. Velocity of flame draws in secondary air to keep the nozzle cool. Units can burn low pressure gas from 400 Btu up, at 3 to 11 in. water column. Each size of assembly is available in four variations of control.



SIZES AND CAPACITIES—Available in eight capacities, ranging from 400,000 Btu per hr rated at 8 boiler hp with a 1½-in. gas inlet and 1/3-hp blower; largest has a rating of 3,300,000 Btu per hr, 66 boiler hp, a 3-in. gas inlet and a 3-hp blower.
MADE BY—*Bryant Heater Co., 1020 London Rd., Cleveland 10, O.*....**573**

Air Register

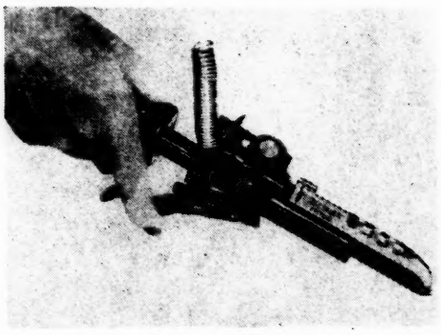
NAME—Waterloo airfoil louver.
PURPOSE—For handling large air quantities at low air resistance.
FEATURES—Airfoil louver is claimed to handle large air quantites at a low air resistance loss. Company also makes a matching return air grille and volume control damper. Blades are inserted and tensioned in frame to assure uniform tension and the prevention of locking of adjustable louvers.



MADE BY — *Titus Manufacturing Corp., P. O. Box 591, Waterloo, Iowa.* **574**

Solder Feed

NAME—Solder-Matic.
PURPOSE — Automatic feeding of solder to soldering iron.
FEATURES — Device takes solder from 1/16 to 3/16-in. diameter and feeds up to 3/16 in. per stroke. Screw adjustment of the steel nozzle guides solder exactly where needed regardless of shape of soldering tip being used.



MADE BY—*Nelpin Manufacturing Co., 45-17 Davis St., Long Island City 1, N. Y.* **575**

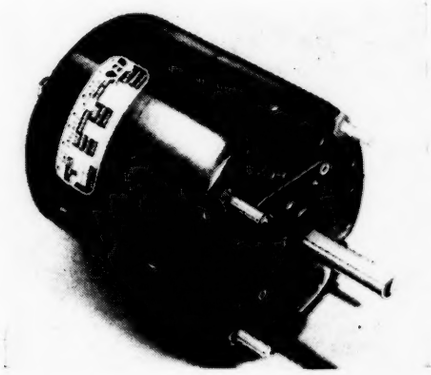
Unit Heater

NAME—Kennard unit heater.
PURPOSE—For space heating and ventilation.
FEATURES—Motor is mounted on the combination bracket and fan guard in such a way as to absorb vibration. Motor is available for single speed and multi-speed operation, single phase, poly phase or direct current with a large choice of electrical characteristics. The heating coil floats in the heater to eliminate stress and strain due to expansion and contraction. Coils are designed to withstand pressures up to 150 lb per sq in. Aluminum fins are spaced six to the inch by means of deep, drawn, contact collars. Seamless copper tubing is used for both condensing tubes and header. Air flow can be directed at any downward angle through adjustable louvers.

LITERATURE AVAILABLE — Bulletin No. 4710
MADE BY — *Kennard Corporation, 1819 S. Hanley Rd., St. Louis 17, Mo.* **576**

Small Motor

NAME—Redmond type W fractional hp motor.
PURPOSE — Small motor designed for fractional horsepower uses.
FEATURES—Motor is of ventilated, 4-pole shaded pole design for a-c operation and is said to operate without interference from radio. Oil is circulated through porous bronze bearings. Motor is micro-balanced and has a machined surface.
SIZES AND CAPACITIES—1/15 hp operates at 1500 rpm on 115 volts, 60 cycle current.
MADE BY — *Redmond Co., Inc., Owosso, Mich.* **577**

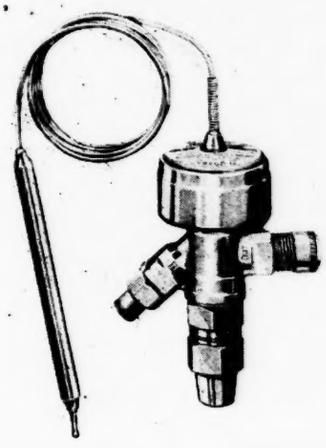


Refrigeration Valve

NAME—A-P thermostatic expansion valve.

PURPOSE—Valve for refrigeration use where a pressure limiting feature is desired.

FEATURES—Diaphragm unit limits the operating suction pressure to a



predetermined setting. Should the pressure build up to a point approaching overload, the diaphragm actuates the needle pin to permit the valve needle to move towards the closed position until the overload condition has passed. Model 211 is non-adjustable but Model 212 may be used in place of any standard thermostatic expansion valve of similar capacity. Types are available to limit pressures to 15, 40, and 55 lb. and for use with Freon 12, and 10 and 30 Methyl.

SIZES AND CAPACITIES—Capacity of models are ½, 1 and 1½ tons Freon; and ½, 1 and 3 tons Methyl.

MADE BY—*Automatic Products Co., 2450 No. 32nd St., Milwaukee 10, Wis.* **578**

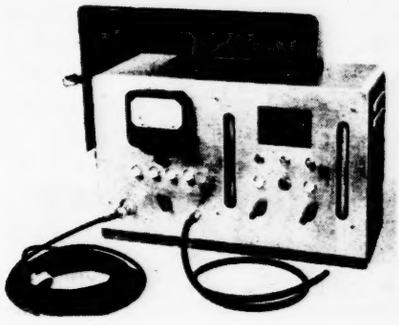
Gas Analyzer

NAME—Micro-gas analyzer.

PURPOSE — For determining and analyzing the concentration of gases or vapors in their toxic range.

FEATURES—Unit works on the principle of electrical conductivity of solutions. Any gas which will ionize in water, with or without heat, may be analyzed by this instrument. Readings may be made directly and continuously by minor adjustments. Unit is portable and weighs about 33 lb.

OPERATION—Gas is drawn into the analyzer by a small pump and this gas passes through a flowrator and then enters a combustion furnace where it is decomposed. From the furnace the gas passes into the cell through which the solution is being pumped. As the decomposed gas dissolves in water the electrical conductivity changes and this is measured on a meter. After the solution passes by the cell it is pumped into a special unit which de-ionizes this solution and returns it to the cell in its original state.



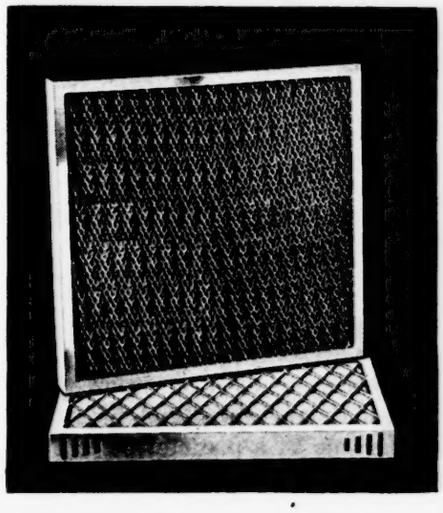
MADE BY—*Davis Emergency Equipment Co., Inc., 45 Halleck St., Newark, N. J.* **579**

Air Filter

NAME—Aircor.

PURPOSE — Permanent, cleanable type air filter.

FEATURES—Galvanized, crimped, expanded metal screen deflects air stream and serves as a lint arrestor. A heavy gage galvanized steel



frame that surrounds the filter has full welded corners and drain slots for cleaning. Filter comes in standard 1 and 2-in. widths. Makers claim a rated filter efficiency of 98.5%

MADE BY—*Air Filter Corporation, 2514-H West Lisbon Ave., Milwaukee 5, Wis.* **580**

Dust Collector

NAME—Mikro-collector

PURPOSE—Dust collection from industrial exhaust systems.

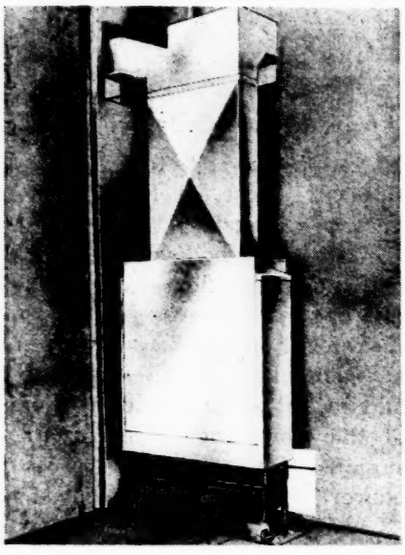
FEATURES—A hollow, annual blow ring hugs and flexes a cylindrical felt bag, as it moves up and down the height of the bag by a chain and sprocket drive. Through a narrow slit around inner periphery of the blow ring, a jet of air is supplied by a positive pressure blower. This counter current of air

is conveyed to the blow ring by a flexible hose to assure a clean felt bag, so that there is a large effective felt surface for cleaning. Makers claim that the installations show an average filter rate of 20 cu ft per sq ft of filter surface per minute. The felt bag is centered in an aluminum housing. Dust laden air is admitted through an opening in the roof of the cabinet and is swept against the inner surface of the bag. Flexing of the bag by blow ring prevents the formation of heavy dust layers against the felt.

MADE BY—*Pulverizing Machinery Co., 194 Chatham Rd., Summit, N. J.* **581**

Warm Air Heater

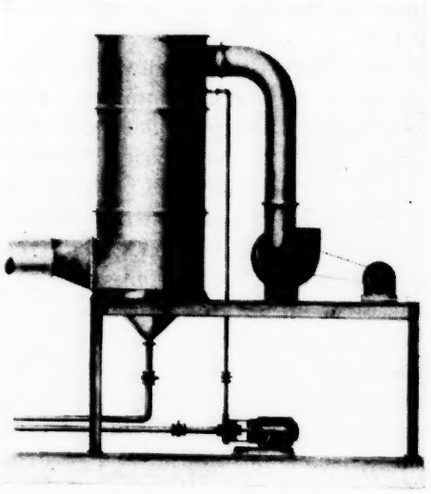
NAME—Royal Jet-Flow heater.
PURPOSE—Space heating for low cost housing.
FEATURES—Furnace burns natural or manufactured gas and is equipped with safety locks on



burner and pilot valves. Fire box is constructed of low alloy, high tensile steel. Without blowers, makers claim that warm air leaves unit at velocity of about 250 fpm. Cold air is drawn in through register at floor level, passes over the fire box, is compressed as it enters a cone-shaped duct to outlet registers at ceiling height. Unit does not require a basement for installation and parts are accessible for cleaning. Three-way adjustable baffle permits heat to be distributed in any desired volume or direction.
MADE BY—*Royal Heaters, Inc., 1024 Westminster Ave., Alhambra, Calif.* **582**

Dust Elimination

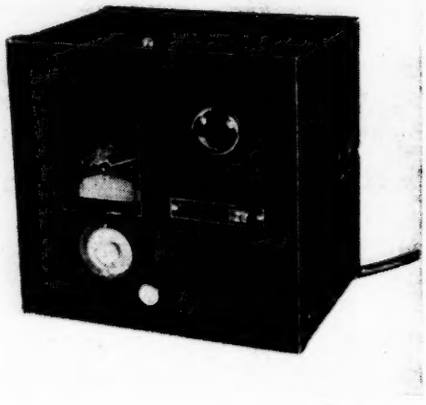
NAME—Pemasco system of dust elimination.
PURPOSE—Removal of dust from exhaust air.
FEATURES—Sprays of liquid inside a dust collecting chamber saturate the incoming air stream and transforms the dust into a dilute sludge which drains from the chamber into settling ponds or tanks. One or more beds of ceramic material are used to assure that dust will be



washed from the air stream. System is said to have great value where complete recovery of fines in exhaust is desired. Wash liquid is recirculated. Float valve admits fresh liquid to make up for loss through evaporation. Unit is fabricated of steel, stainless steel, monel and other metals.
SIZES AND CAPACITIES—In sizes ranging from 1,000 to 300,000 cfm.
LITERATURE AVAILABLE — Bulletin A-10.
MADE BY—*Paul E. McKamy & Associates, 8 S. Dearborn St., Chicago 3, Ill.* **583**

Time Switch

NAME—Palo time switch Senior.
PURPOSE—Self-contained control.
FEATURES—Unit has the necessary



equipment to turn on or off many types of laboratory and industrial apparatus on a predetermined time schedule. It operates at 110 volts, a-c. The clock, of conventional design, is self-starting, with a sweep second hand. A switch dial, geared to the clock mechanism, revolves at the rate of one revolution per day. Dial face has two adjustable switch fingers, one for on and the other for off. This operates a relay of the double pole type, capable of breaking 20 amps of non-inductive or heater load. A pilot light on the front of the case indicates condition of the relay.
LITERATURE AVAILABLE — Descriptive bulletin.
MADE BY — *Palo-Myers, Inc., 81 Reade St., New York 7, N. Y.*.....**584**

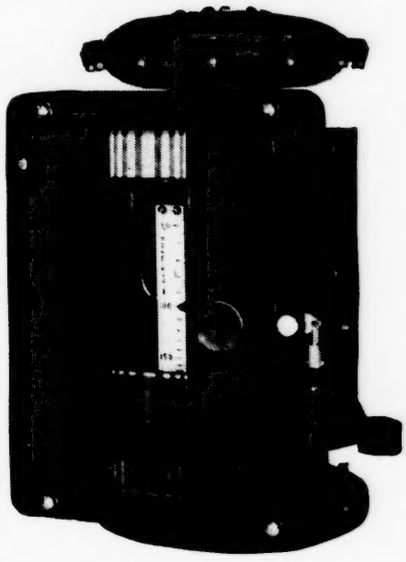
Radiator Valve

NAME—Heat timer valve.
PURPOSE—For room temperature control with steam radiators.

FEATURES — Valve is installed in place of customary air valve. On cylindrical side of unit is a scale of temperatures to which the timer valve can be set. Movable indicator can be rotated until it points to the desired temperature on the scale. Valve will automatically discharge air only when temperature within the room falls below indicated setting. This permits steam to enter the radiator until desired room temperature has been reached. At that point thermostat within the valve keeps the vent closed. No steam feeds into the radiator when no heat is wanted.
MADE BY—*Heat Timer Corp., 160 Fifth Ave., New York, N. Y.*.....**585**

Air-Fuel Control

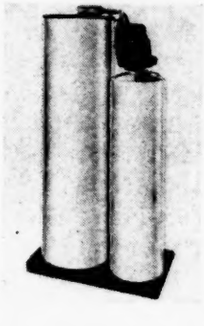
NAME—Air-fuel ratio controller.
PURPOSE—To control and regulate air-fuel ratio in large industrial furnaces.



FEATURES—A rugged, balance-type instrument said to have ample flexibility to meet any operating condition. Its ratio can be adjusted manually for operation with fuels of varying Btu content. As fuel flow changes, instrument maintains air-fuel ratio constant at the desired control point. Or, it can provide automatic variation, increasing or decreasing the ratio at reduced fuel flow rates. Controller employs electric motored valve drives.
MADE BY—*Leeds and Northrup Co., 4934 Stenton Ave., Philadelphia 44, Pa.* **586**

Water Softener

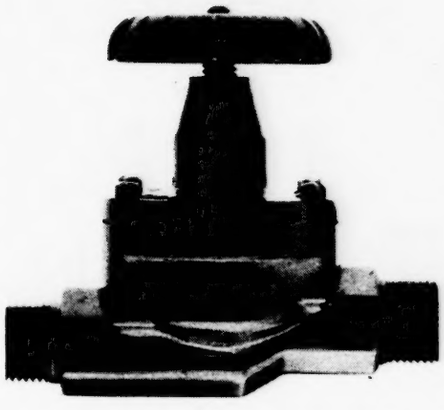
NAME — Pepco automatic water softener.
PURPOSE—Water softening unit designed for small commercial installations.
FEATURES—Device operates on the principle of using zeolite for automatically removing hardness from water. There is a complete zeolite regeneration cycle every 24 hours. The only attention required is the customary replacement of common rock salt which is used in the regeneration process. Entire process is controlled automatically through a time-clock mechanism.



MADE BY—*Auto-Softener Distributing Co., 21137 Grand River Ave., Detroit 19, Mich.* **587**

Valves

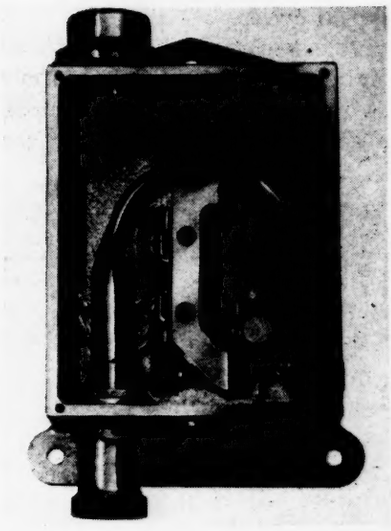
NAME—Airco station outlet valve.
PURPOSE—Combined shut off and check valve for use on drops or risers of a piping system.
FEATURES — Valve is a packless diaphragm type unit. As a check valve, it permits the flow of gas in one direction only. As a shut off valve it permits repairs or alterations without disturbing other stations in the line. Body, bonnet and handwheel are brass forgings.



MADE BY — *Air Reduction Sales Company, 60 East 42nd Street, New York 17, N. Y.* **588**

Pressure Switch

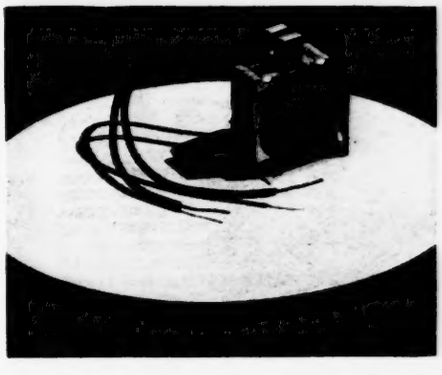
NAME—Meletron pressure operated switch-model 310.
PURPOSE—Switch is controlled by pressure variations.
FEATURES—Bourdon tube pressure response element permits adjustments to be made with comparable pressure gauge accuracy. Available in several combinations of switching and adjusting arrangements depending on service required as: Single pole, double throw switch with standard adjusting screw for application where few pressure settings are required; manual reset type where circuit is not to be reenergized until original cause of switch action is located; external



adjustment knob with locking arrangement where frequent pressure change settings are required.
MADE BY—*Meletron Corp., 950 N. Highland Ave., Los Angeles 38, Calif.* **589**

Solenoid

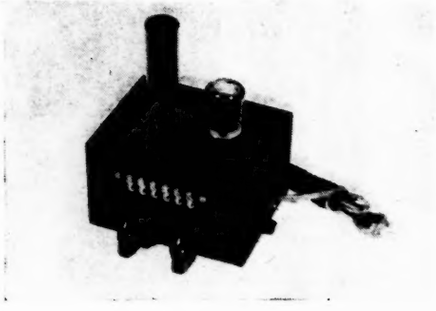
NAME—G. E. all welded solenoid.
PURPOSE—For providing strong pull for trip mechanisms and safety devices.
FEATURES — Solenoid operates on 110 volts, 60 cycles and develops a maximum pull of 0.26 lb in a ½ in. stroke. The L-shape mounting bracket permits horizontal or vertical mounting. Frame and bracket



are welded together; frame laminations are also welded. Pole shader is brazed into place to provide quiet operation.
MADE BY — *General Electric Co., Schenectady 5, N. Y.* **590**

Cycle Timer

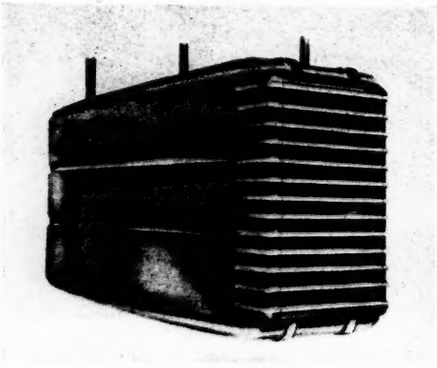
NAME—Wilson repeat cycle timer.
PURPOSE—Automatic timing for industrial cycles.
FEATURES—On and off periods ranging from 0.1 second to 4 minutes are available. Instrument can be adjusted. Makers claim that



wide variations in line voltage will have less than 1% effect on the timing intervals. Relay contacts having up to two transfers each rated at 10 amp at 115 volts can be obtained. Timer is housed in a standard chassis 3 x 4 x 5 in.
MADE BY—*G. C. Wilson & Co., 2 N. Passaic Ave., Chatham, N. J.*.....**591**

Suspended Heater

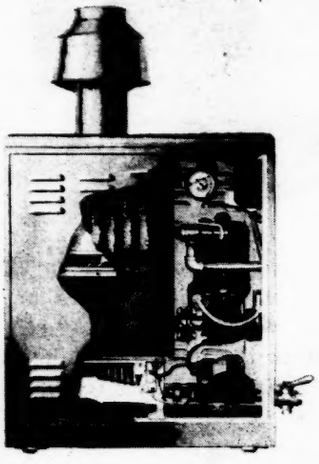
NAME—Pacific suspended heater.
PURPOSE—Space heating.
FEATURES—Heavy duty fan forces air over thermo-dynamically shaper sections. Products of combustion are exhausted through a vent pipe.



Adjustable louvers direct warm air flow through polished stainless steel grilles. Unit has been designed to burn natural, mixed, or manufactured gas. Standard equipment includes low voltage type solenoid gas valve, secondary transformer thermostat, automatic pilot, limit control, pilot valve, manual shut-off valve and low pressure gas regulator.
SIZES AND CAPACITIES—Five sized units ranging in Btu per hour output from 52,800 to 160,000.
MADE BY—*Pacific Heater Div., Naco Mfg. Co., P.O. Box 310. Huntington Park, Calif.* **592**

Gas-Fired Boiler

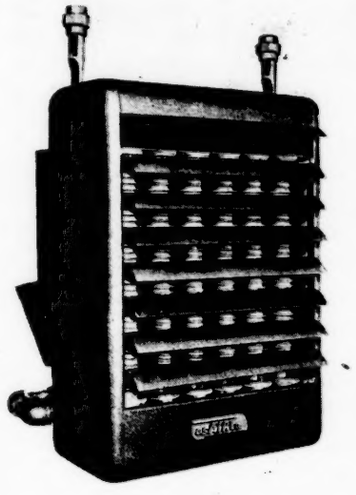
NAME—Kehm Master gas-fired hot water heating boiler.
PURPOSE—Space heater using gas as fuel.
FEATURES—Packaged unit contains all accessories necessary for a complete forced hot water heating system including circulating pump,



relief valve, limit control, relay, gage, pilostat, draft diverter, and is completely wired and assembled ready to operate. Unit has an adjustable flame with a bunsen type burner which permits gas input to match the amount of radiation installed. Heat exchanger is composed of cast iron sections and Thermek spine tubes through which gas is filtered.
SIZES AND CAPACITIES—Two sizes, one of 80,000 and the other, 120,000 Btu output.
MADE BY—*The Kehm Corporation, 135 South LaSalle Street, Chicago, Ill.* **593**

Unit Heater

NAME—Model CS-94 gas-fired unit heater.
PURPOSE—Space heating with gas as fuel.
FEATURES—Unit has cast iron ribbon type burners that can be easily removed for cleaning, and cast iron heat exchangers with internal as well as external fins. A fan with 16-in. kidney-shaped blades



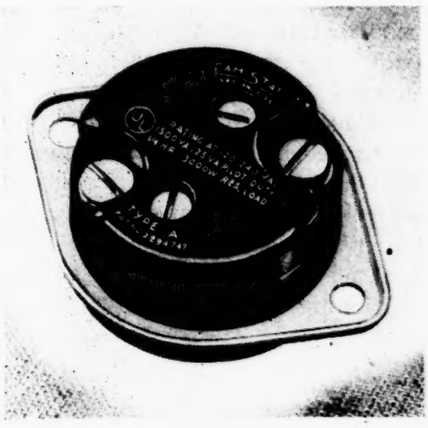
was designed for quiet operation. Unit is provided with automatic controls.
MADE BY—*Great National Air Conditioning Corp., Dallas, Texas, holders of exclusive manufacturing license from United States Air Conditioning Corp.* **594**

Indoor-Outdoor Thermometer

NAME—Marsh duo-temp.
PURPOSE—Single instrument for measuring both inside and outside temperatures.
FEATURES—Face of instrument is divided into two separate temperature ranges. Top range indicates the outdoor temperature while the bottom shows the inside temperature. The Bourdon principle is utilized in the design of this temperature recording unit. Tubing runs from thermometer to bulb placed outside for outdoor readings.
MADE BY—*Jas. P. Marsh Corp., 2073 Southport Ave., Chicago 14, Ill.* **595**

Temperature Control

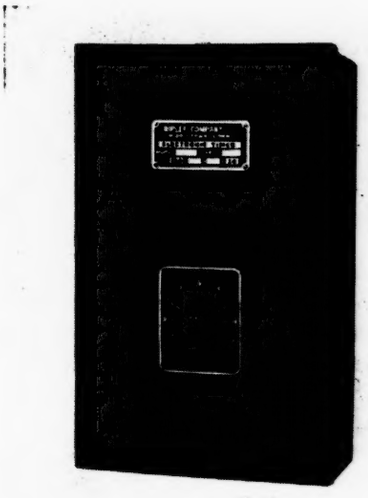
NAME—Cam-Stat Type A control.
PURPOSE—Temperature control for low-voltage applications.
FEATURES—Temperature ranges ad-



justable from 50 to 350F or factory set to designated specifications. Switching arrangements are SPST break or make on temperature rise, or SPDT and independent circuit double throw. Unit is 1-9/16 in. in diameter and 1-13/32 in. deep. It has been especially designed as a high limit control on unit heaters and forced air furnaces.
MADE BY—*Cam-Stat, Inc., 2310 So. La Cienega Blvd., Los Angeles 34, Calif.* **596**

Welding Timer

NAME—No. 52D Timatron.
PURPOSE—Electric welding timer for spot welding.
FEATURES—Unit can time any machine up to 10 kva and is adjustable from 1 to 5 seconds by a hand control. Once set for a particular thickness of metal, time becomes automatic for the entire production run.
MADE BY—*Ripley Co., Inc., Middletown, Conn.* **597**

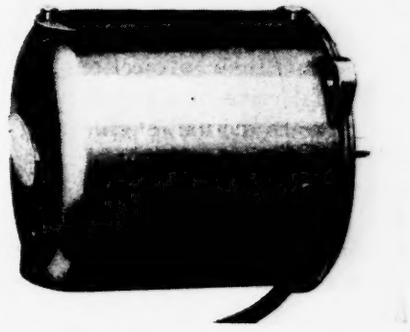


Oil Burner

NAME — HomEase TMO-A oil burner.
PURPOSE—Small size oil burner for heating homes.
FEATURES—Burner has capacity of ¾ to 1.5 gal per hr and is equipped with a device that regulates the size and shape of the fire-box. Special device provides additional air when starting. Oil and air mixture automatically changes with operating conditions.
MADE BY—*HomEase Products Div.; Bogue Industries, Paterson, N. J.* **598**

Motor

NAME—Jack & Heintz fractional horsepower motor.
PURPOSE—A specially designed electric motor for oil burner applications.
FEATURES—Motor is 1/6 hp split phase type, designed to operate on



115 volts at 60 cycles, with a constant speed of 1725 rpm. A manual reset overload protector is built in for protection against harmful overloads. Starting current 18 amps and full load current 3.1 amps.
MADE BY—*Jack & Heintz Precision Industries, Incorporated, Cleveland 1, Ohio.* **599**

NOVEMBER, 1947

TO OBTAIN FURTHER INFORMATION ON NEW EQUIPMENT

On the list below, circle the item number of the equipment in which you are interested, using the number found at the end of each item.

563	564	565	566	567	568	569	570
571	572	573	574	575	576	577	578
579	580	581	582	583	584	585	586
587	588	589	590	591	592	593	594
595	596	597	598	599			

Print your name and address, detach and mail to
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Title (Must be shown).....

Firm (Must be shown).....
(For prompt service, title and firm name must be shown above.)

Business Address .....

City.....**Zone**.....**State**.....**11/47**

Abstracts AND Reviews

Publications abstracted in this department should be ordered direct from publisher.

DESCRIPTIVE GEOMETRY

"Practical Descriptive Geometry" is a step-by-step development of a difficult subject by one of its practicing teachers. S. E. Rusinoff, M. E., now associate professor of mechanical engineering at Illinois Institute of Technology, makes use of his wide industrial experience with Westinghouse, Borg Warner, and others, in developing the basic fundamentals of draftsmanship through the geometry of lines, planes, and curved surfaces to the accurate drawing of these elements and their projections and intersections. The prime objective of each discussion is its ultimate application to actual engineering and drafting room problems. The book is profusely illustrated with carefully drawn examples. An appendix includes mathematical short cuts and tables.

Practical Descriptive Geometry, by S. E. Rusinoff. Cloth bound, 8½ x 6 in., 260 pages. Published by American Technical Society, Drexel Ave. at 58th St., Chicago 37, Ill.

• • •

TUBE FITTERS' MANUAL—A 76-page manual which by means of photographs and drawings presents information for men concerned with tubing for fluid handling circuits. It covers the subject of tubing and fittings, use of the tube fitter's tools, planning the tubing circuit, measuring the line, and helpful hints on tubing systems. It describes the proper bending of tubing. The Parker Appliance Co., Cleveland 12, Ohio.

FUEL SAVINGS—In a single publication, the Bureau of Mines has combined all the material on fuel conservation that was successfully used during the war years. The most effective material distributed during the two-year campaign were quiz sheets covering nearly every phase of fuel savings. All of these question and answer sheets are included in Bulletin 466. A Guide for Reducing Fuel Consumption in Commercial Plants. Price, 50 cents. Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C.

PIPE COILS AND FIN COILS—Data book of 34 pages covers the fundamentals of pipe and fin coil calculations for draftsmen and designers. It gives in detail the methods of designing pipe and fin coils for heating and cooling loads. It also has information on inspecting, testing and finishing coils. Price, $1.50. Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.

WARM AIR HEATING DESIGN—Typed instructions and tables used by the Technical Education Committee of the National Warm Air Heating and Air Conditioning Association at Michigan State College for the society's short course. This manual No. 9 is the design procedure for buildings of more than 120,000 Btu per hour heat loss. Together with this manual are included three work sheets—heat loss sheet; warm air and return air branch, stack and register sizing sheet; and warm air and return air trunk duct sizing sheet. Classroom Edition of manual, 50 cents. National Warm Air Heating and Air Conditioning Association, 143 Public Square, Cleveland 14, Ohio.

COAL AND COKE STANDARDS—The 1947 ASTM Standards on Coal and Coke covers some 33 specifications, methods of tests and other data developed by the association. It also contains eight proposed methods which are published solely for information and comment. Included are 14 methods of testing coal covering sampling, tests for size, dust, etc.; specifications covering classification of coals by rank, by grade, and for gas and coking coals. Publication contains 162 pages, paper bound, price, $2. American Society for Testing Materials, 1916 Race St., Philadelphia 3, Pa.

LOCOMOTIVE SMOKE ABATEMENT—The use of overfire air jets for locomotive smoke abatement is covered in a booklet by Eugene D. Benton, assistant supervisor, Fuels Division, Batelle Memorial Institute on "Overfire Jets and Controls for Locomotive Smoke Abatement." Recommendations contained in the report are the result of engineering research on jet performance and experience gained as the result of operating many classes of locomotives. A large number of illustrations and tables are included in this 14-page booklet. Copies, 25c each may be obtained from Bituminous Coal Research, Inc., 912 Oliver Bldg., Pittsburgh 22, Pa.

AIR CONDITIONING APPLICATIONS—In a 16-page booklet, Application Engineering Standards for Air Conditioning for Comfort, are contained text and tables covering design load factors, design inside and outside conditions, sunlight radiation through glass, ventilation and infiltration, transmission coefficients and air distribution. A simple cooling load calculation form is included in the section on load calculations. There is also a section on heating loads and a short form for heating load calculations. Price, $1. Air Conditioning & Refrigerating Machinery Association, Southern Building, Washington, D. C.

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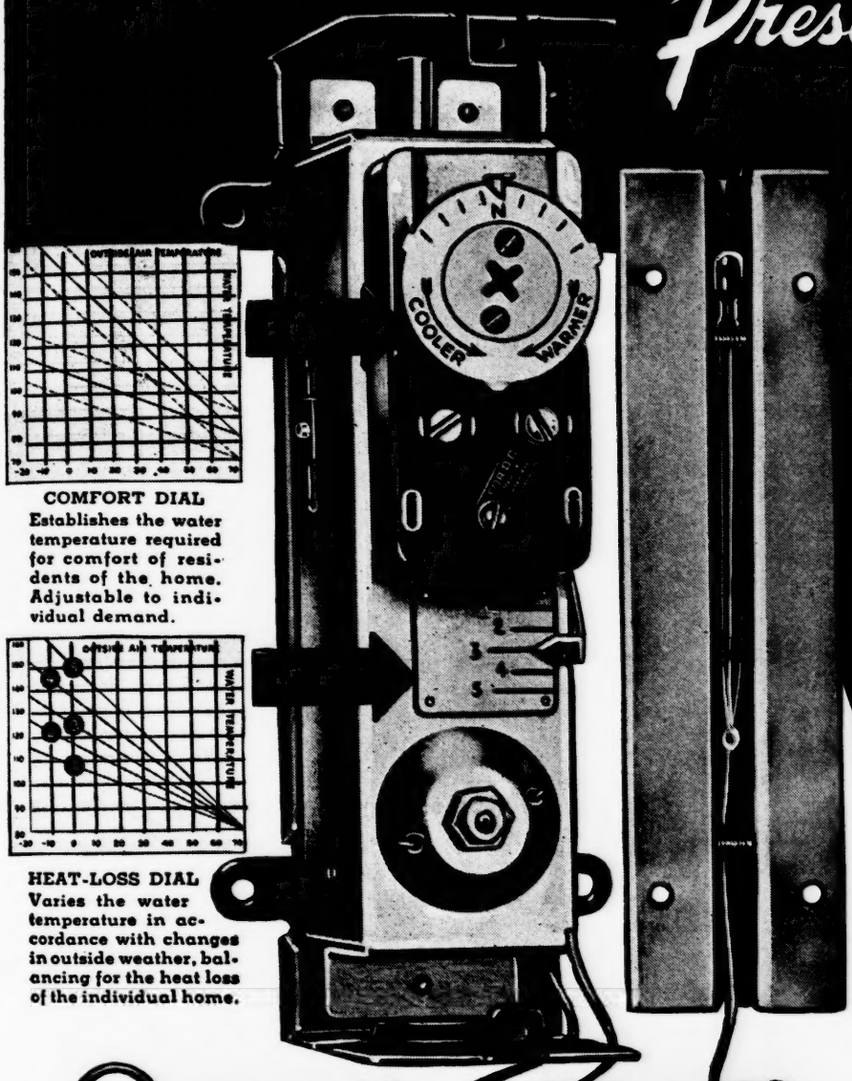

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NEWS OF THE MONTH

PERFORMANCE STANDARDS

for integral horsepower electric motors crown 40 years of development by NEMA.

New performance standards for industrial type integral horsepower three-phase squirrel-cage induction motors of 1 to 200 hp have been announced by the National Electrical Manufacturers Association.

These standards, are of unusual significance to the industrial motor user, according to F. J. Geiger, Allis-Chalmers Mfg. Co., who discussed them with press representatives in New York City on October 30 in his capacity as vice-chairman of the Integral Horsepower Subsection of the NEMA Motor and Generator Section. They provide, he said, a sound basis for selecting the right motor for any industrial application, simplify the specification of minimum acceptable limits of operating performance, and assure interchangeability of motors made by different manufacturers.

• **DEVELOPMENT.**—Standardization in the electric motor industry, Mr. Geiger noted, started 40 years ago, and has resulted in the development of standards relating to such matters as preferred horsepower and speed classifications, mounting and frame dimensions, enclosures for various locations and atmospheric conditions, and insulation types and associated limits on ambient temperature, temperature rise, and loading.

The new performance standards which will become effective January 1, 1948, define five basic types of polyphase induction motors, designated Design *A*, *B*, *C*, *D*, and *F*, each of which offers a different combination of torque, speed, and current characteristics to meet the operating requirements of various industrial applications. The recently retired chairman of the Motor and Generator Section, C. P. Potter, Wagner Electric Corp., St. Louis, described the five motor types, while M. S. Hancock, Westinghouse Electric Corporation, who is vice-chairman of the Section's General Engineering Committee, discussed the performance characteristics which enter into the definition of these types—starting and breakdown torques, locked-rotor current, and slip—and the differences in design and materials used to obtain various values for each.

General background on the electric motor business was supplied by E. E. Helm, Reliance Electric and Engineering Co., Cleveland, newly elected chairman of the Motor and Generator Section.

• **DESIGNS.**—The squirrel-cage induction motor, said Mr. Hancock, is the most reliable and lowest cost motor which can be built. A variety of designs has been developed to meet different industrial application requirements. Most popular of these types are the Designs *B* and *C*. Design *B* has normal starting torque adequate for a wide variety of industrial machines and drives and a starting current usually acceptable on power systems. The Design *C* motor has higher starting ability for such applications as loaded compressors, conveyors, and the like. Design *A*, however, provides exceptionally high breakdown torques, but at the expense of high locked-rotor currents which normally require provision for starting with reduced voltage. Design *F* motors, on the other hand, have exceptionally low values for both characteristics.

Design *D* uses a high-resistance type rotor to obtain variation of speed with load, desirable for some applications with rapidly fluctuating loads. This type also gives extremely high starting torques with moderate starting currents. The relatively constant speeds under varying load conditions required in most applications are provided in the other four designs, which have slip values of less than 5%.

It was pointed out that the five motor types do not represent new departures in design, but rather the application of standard nomenclature to accepted practice in the electric motor industry, which in turn is realistically based on the established requirements of industry.

• **VOLUNTARY.**—It was also emphasized that each motor manufacturer is free to build motors to NEMA standards or not as he sees fit. Experience has indicated, however, that the advantages of standardization to the purchaser and user result in volume demand for standard items which most manufacturers are eager to supply.

UNIFORM BUILDING CODES

advanced as Building Officials Conference offers prototype code and Federal agency issues guide to performance standards.

As the Building Officials Conference of America ended its four-day annual meeting in Columbus, Ohio, it was announced that members of the construction and associated manufacturing industries would be canvassed for suggestions regarding a basic building code developed by the Conference.

• **WHAT TO DO.**—The first part of the code, now nearing completion, is termed the "What To Do" part by George E. Strehan of New York City, correlator of the code. It is composed of minimum performance standards for structural, fire-resistance, and sanitary safety.

After two or three months required for preparation of the code and distribution to interested members of the construction industry, 90 days will be allowed for the offering of suggestions, and the document will then be put in final form for submission to local communities.

This performance phase of the code will be supplemented later by a section of specific methods and

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Easy Bending	Revere Copper Water Tube is readily bent by hand. Simple wood templates provide proper radius and spacing.
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In radiant panel heating, just as in other types of heating systems, *trouble always costs more than Revere Copper.* You can specify or install, in addition to Revere Copper Water Tube, such other long-lived Revere materials as Red-Brass Pipe; Sheet Copper for tanks, ducts, pans and trays; Dryseal Copper Refrigeration Tube (dehydrated and sealed); Copper oil burner, heat control and capillary tubes.

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materials. A third code will be developed for very small communities.

• **PERFORMANCE STANDARDS.**—A uniform basis for measuring the adequacy of building materials and home construction methods is offered in a new publication, *Performance Standards,* issued by the Technical Office of the Housing and Home Finance Agency. The pamphlet proposes performance standards for structural elements of the house and is supplemented by maps indicating wind velocity pressure and snow load in various sections of the United States. The document was prepared by the Building Regulations and Construction Standards Branch of the Technical Office serving the Office of the Housing Expeditor of the National Housing Agency in consultation with the National Bureau of Standards, and other Federal agencies, technical and engineering societies, trade associations, and professional engineers. Several drafts were circulated for comment and criticism prior to publication in its present form.

SMOKE LAW

has smoggy debut in Pittsburgh. Shortages of smokeless fuel and equipment blamed.

Pittsburghers are discovering that it takes something more potent than a city ordinance to dissipate their famous pall of smoke according to a *Business Week* report. After a century of agitation, a city ordinance effective Oct. 1 solemnly banned the city's smog.

The following week eager esthetes admitted that reforms must bow, at least temporarily, to economics. In this case it was a shortage of smokeless fuel, lack of smoke-abating equipment. The esthetes' discomfiture is enjoyed by laundrymen and by old-time steel makers who are wont to judge the rate of industrial activity by the thickness of the atmosphere.

• **SMOGGY DEBUT.**—Pittsburgh's official bid for blue skies follows similar moves by St. Louis and Cincinnati. Ironically, the steel city's introduction to the new law last week was subdued by a thick natural fog. It was the same on the two following days.

But the gradual achievement of blue skies will be speeded when surrounding Allegheny County invokes a code that backs up the city ordinance.

The new code requires that within city limits only smokeless fuels may be burned—except where stokers are used to control the gas and soot thrown off by high-volatile coal. Dealers are now prohibited from delivering high-volatile bituminous to homes, apartments, and other buildings that are not equipped for smokeless combustion. Big buildings, railroads, industries had already come under the ordinance. Chimneys of the city's 150,000 dwelling houses were last to be affected.

• **ENFORCEMENT TROUBLES.**—Early difficulties indicate that the 13 smoke inspectors are a woefully inadequate force. They backed predictions made by opponents of the law. John Lewis' United Mine Workers and allied interests contended that there wasn't enough smokeless coal in the Pittsburgh district, that the banned types of coal were cheaper, that smokeless stokers were scarce. Coal dealers currently report a shortage of acceptable coal in the city, predict a black market in the fuel.

Under the circumstances, enforcement authorities promise to be lenient. Mayor David L. Lawrence has appointed a committee to study complaints. He is empowered to waive the restrictions for six months in an emergency. Enemies of the law are expected to take full advantage of loopholes.

Certain consumers can make smoke for stated intervals. Thus a locomotive fireman can pollute the air for one minute each hour. Industrial plants may indulge for longer periods while cleaning or starting fires. Some groups are "excused" from penalties while smoke-eliminating equipment is installed.

• **OTHERS TROUBLED.**—While Pittsburghers coughed and argued, Cincinnatians had similar troubles. Cincinnati's antismoke ordinance, passed last May, was blamed for much of the shivers during a recent cold snap.

A committee of the city council heard coal dealers protest this week that teeth will continue to chatter unless the ban against smoky coal is called off for a time. Jesse D. Locker, committee chairman, announced that he intended to recommend a suspension. Dealers contended that immediate action would favor the acquisition of high-volatile coal in competition with other cities.

• **THE PIONEER.**—St. Louis, a pioneer in smoke control, is able to watch the gropings of fellow municipalities with sympathy. Its burghers consider their antismoke campaign an established success. They speak of 1939's "black Tuesday" (when daylight turned pitch black) as a healthy man speaks of past afflictions.

St. Louis' antismoke ordinance took effect during the winter of 1940-41. From Oct. 1, 1940, to Oct. 1, 1941, the hours of dense smoke totaled 197. The score for the uncontrolled October-to-October period of 1939-1940 were 716 gloomy hours. For the first year of control, infractions of the antismoke ordinance numbered 148; last year there were seven.

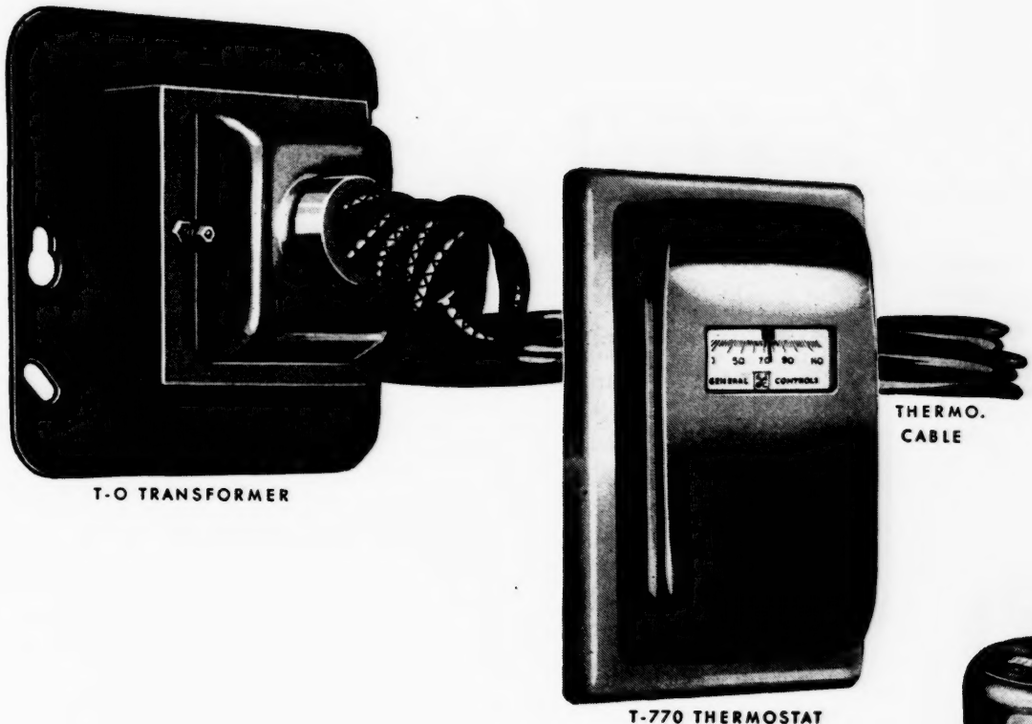
SMOKELESS FURNACE

developed for burning bituminous coal in a warm air heating unit.

A smokeless magazine type warm air furnace burning bituminous coal has been developed by Bituminous Coal Research, Inc. for residential heating. With research completed, and application made for patents, the institute is now trying to interest qualified manufacturers in the production of this unit on a nominal royalty basis. Commercial designing for mass production will be undertaken by the manufacturer.

• **CAPACITY.**—Rating tests over a 12-hour firing cycle gave an output of about 80,000 Btu per hr at an over-

T-23 OIL HEATING PACKAGE SET



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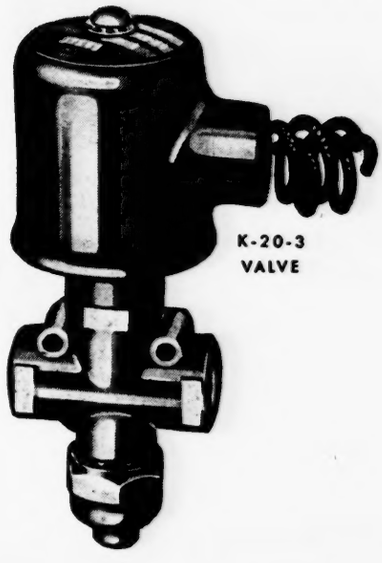

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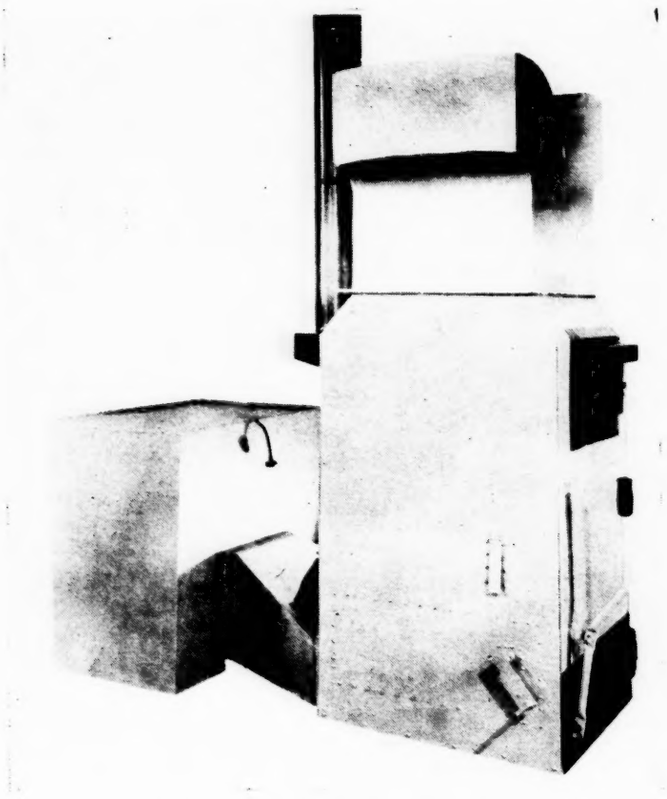


Smokeless bituminous coal furnace

all efficiency of better than 60%. Tests were conducted with two types of bituminous coals and within the limitations of Commercial Standards CS-109-44. Both coking and free-burning bituminous coals can be used. At full output the magazine can hold about a 12-hour supply.

Natural draft from an average chimney is said to be sufficient to operate the furnace. Automatic draft dampers can be applied to the furnace to regulate the heat release.

Ashes are removed from the fuel bed by a special type shaking grate. Slate and occasional clinkers are removed by a dumping grate section.

TWO ANTHRATUBE MODELS

using new coal burning principle are now in production.

Two production models of the anthratube, a new heating unit which is said to deliver coal savings of 15 to 38% in home installations, were shown to the general public for the first time last month by Anthracite Institute, which developed the basic principle. Frank W. Earnest, Jr., president of the Institute, announced that the anthratube is being produced by several manufacturers and is available for home installation.

• **PRINCIPLE.**—"The anthratube is a complete boiler-burner unit, based on an entirely new coal-burning principle, which provides completely automatic heat for the home with greater economy, cleanliness and convenience than has ever been considered possible with conventional methods of burning anthracite," said Mr. Earnest. "Since the principle was announced by the Institute three years ago it has been subjected to intensive research and widespread testing, including year-'round operation in 150 homes. In these home installations, under conditions of actual use, it has been shown to operate at 80% or greater efficiency."

In brief, the anthratube restricts the actual burning of coal to a very small area and provides a new controlled feeding method which allows only the amount of coal to be burned at one time into the fire box. The new models, smaller by at least one-third than the usual residential boiler, deposit their ashes into a bin that is easily removed and emptied by the tender. An induced draft fan is said to eliminate the loose fly ash that sometimes escapes from ordinary furnaces. Both units provide hot water as well as steam.

• **MODELS.**—The hopper model, made by the Delaware Lackawanna and Western Coal Co., has a vertical tube and fire box, which is fed manually with chestnut coal about every three days—less often in warmer weather. Ash is deposited in the bin by a moving grate which automatically shears off the ashes as the coal is consumed. A draft keeps the coal gases swirling at high speed against the heat-transmitting surfaces of the boiler.

A bin-feed model made by Axeman-Anderson Associates and American Boiler Works, burns pea-size coal and features a long screw feed to the burner from the coal bin. A draft fan keeps the gases whirling so that more heat is utilized from the coal and separates fly ash from clean gases, depositing it in the ash bin underneath the unit.

Both units are recommended for houses of eight rooms or under. J. E. Axeman estimates that his model can be installed for about $650, while a spokesman for the D. L. and W. Co. said that the hopper model costs $500, exclusive of installation charges.

A boiler-burner unit of 1,000-ft steam capacity, using the anthratube principle in part, has been developed by Motorstoker, Hershey Machine & Foundry Co., Manheim, Pa., and in Canada a "horizontal" type of anthratube is being marketed by Huron Engineering & Research Co., Goderich, Ont.

NEW INSTRUMENTS

for measuring air contamination, demonstrated at New York AIHA meeting.

Four new instruments, three of them now in production, for the measurement of air contamination by halogenated hydrocarbons (carbon tetrachloride, chloroform and many other compounds) were demonstrated at the October meeting of the Metropolitan New York Section, American Industrial Hygiene Association.

DUCTS in "World's Largest Drug Store" INSULATED WITH FIBERGLAS*

The new Rexall Building located in Los Angeles, California, which contains the Administrative Headquarters and the "World's Largest Drug Store". Albert F. Roller, San Francisco—Architect; Louis C. Dunn Co., San Francisco—Contractor.

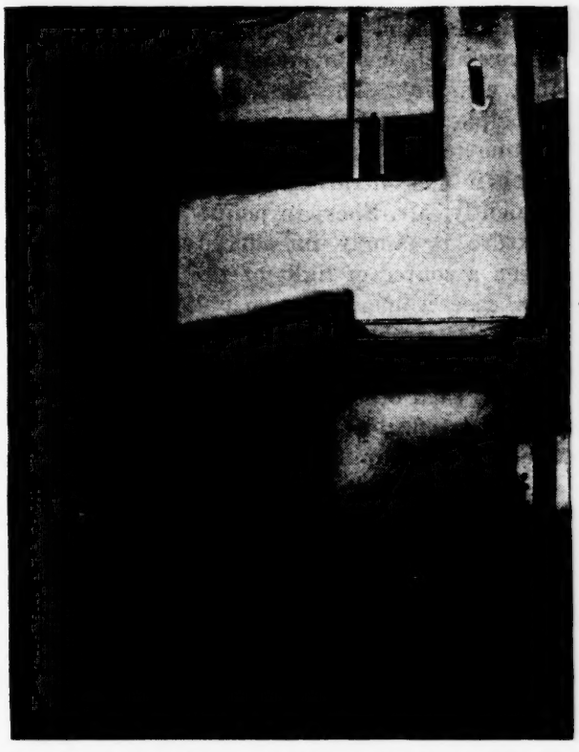
The entire air-conditioning duct system in the new national headquarters of the Rexall Drug Company is insulated with Fiberglas.

In more and more buildings and plants throughout the United States, Fiberglas Thermal Insulation, in various forms, is being used because it possesses a combination of advantages found in no other single material. Fiberglas Insulation is high in thermal insulating efficiency; firesafe; lightweight; offers excellent sound-absorbing qualities; has negligible moisture pickup; will not rot or decay; is odorless and will not hold odors; provides no sustenance for vermin.

Fiberglas Thermal Insulation is available in forms to meet virtually every insulation need in temperature ranges from sub-zero to 1000° F. All can be applied by standard methods.

Be sure that you have all of the facts about the use of Fiberglas for duct insulation and other requirements. Write for booklet "Fiberglas Insulations for Industry". Owens-Corning Fiberglas Corporation, Dept. 912, Toledo 1, Ohio. Branches in principal cities.

In Canada: Fiberglas Canada Ltd., Toronto, Ont.



Exterior of one of blower units showing Fiberglas-insulated ducts surfaced with muslin. Mundet Cork Corporation, Los Angeles, California—Applicator.

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• **DEMONSTRATIONS.**—Dr. Robert Rosenthal, Industrial Instruments, Inc., demonstrated a semi-portable instrument, readings of which are directly convertible to parts of contaminant per million parts of air. In explaining the principle of the instrument, Dr. Rosenthal described it as consisting of a pump for introducing a measured volume of air-vapor mixture; a furnace for raising the temperature of the mixture above the combustion point; a source of distilled water at constant level; a means of mixing streams of distilled water and burned air-vapor mixture at a constant rate; electrodes introduced into the resulting solution and balanced by a Wheatstone bridge arrangement in such a way that variations in conductivity brought about by varying halogen ion concentration are registered on a calibrated dial. The conductivity, Dr. Rosenthal pointed out, is directly proportional to the halogen ion concentration which in turn varies with both the degree of contamination and the number of halogen atoms in a molecule of contaminant.

A smaller instrument manufactured by Davis Emergency Equipment Co. was demonstrated by N. G. White of Merck & Co. This instrument is similar in principle to the larger unit demonstrated by Dr. Rosenthal but readings are made on an ordinary volt meter and are applied to calibration curves drawn for each of the contaminants measurable by the instrument.

• **PHOTO-ELECTRIC.**—A different principle is used in an instrument manufactured by Mine Safety Appliances Co. and demonstrated by John Sherwin of that company. The MSA instrument depends on the principle that halogenated hydrocarbon vapors are opaque to certain frequencies of light in proportion to their concentration in a vapor-air mixture. Consequently, Mr. Sherwin pointed out, the contaminated mixture is simply introduced as an obstruction between a source of light of the required frequency, in this case a mercury vapor lamp, and a system of photo-electric cells. There are two photo-electric cells in the instrument, and these are balanced electrically against each other in uncontaminated air by means of a Wheatstone bridge. Differences in opacity between clear and contaminated air have the effect of unbalancing the system to an extent proportional to the degree of contamination. This unbalancing is measured on a volt meter, reading of which is again applied to calibration curves for determining the number of parts of contaminant per million parts of air.

• **EXPERIMENTAL.**—An instrument still in the experimental stage was demonstrated by C. R. Hendrixon of Chlorine Products Division, Dupont Co. Operation of this instrument depends on the spectroscopic analysis of contaminated air burned in a flame arc between a platinum and a nichrome electrode. A molten drop of copper oxide on the nichrome electrode acts upon the contaminated specimen in such a way that the light frequency characteristic of cupric halide is discernible, and a means of measuring the intensity of this frequency band results in a simultaneous measuring of the degree of contamination.

• **EXTRA.**—In an additional, unscheduled demonstration, William R. Bradley, American Cyanamid Co., exhibited and explained a small, hand instrument for the colorimetric determination of the presence of carbon monoxide. This instrument was developed during the war by the National Bureau of Standards and is being manufactured by Mine Safety Appliances Company.

ANTHRACITE MINERS

assume role of producers in urging measures to combat competition.

Taking the view that the fate of miners is tied up with the business of selling coal, District 1 of the United Mine Workers of America, through its officers, has issued a call to the hard coal diggers to withhold their patronage from those regional businessmen and individuals who have replaced their anthracite-using stoves and furnaces with oil burners. Also, miners have been urged by union leaders to increase man-hour production and reduce absenteeism in an effort to improve the competitive position of anthracite.

• **ECONOMIC LIFE.**—There are approximately 45,000 miners employed in District 1, and mining is the basis of the area's economic life. The action of people living in the anthracite counties switching to oil as a home-heating fuel was termed one of "shortsightedness and ingratitude" because, the mine union officers charged in their report to a recent quadrennial convention, the people who do so "prosper on the money" spent by the miners.

"All members of our union and their families" were then asked, when they spend "their hard-earned money, to remember and spend it with friends of the Industry . . . spend it only at places that are loyal to and burn anthracite coal."

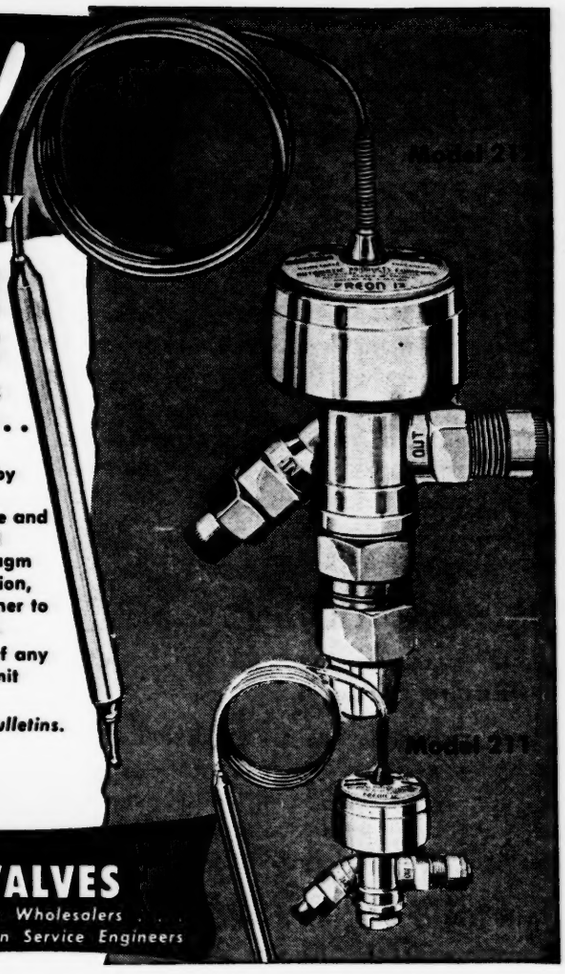
A union list of businesses and individuals in the Wilkes-Barre-Scranton area who are "reported to use oil" includes the names of several doctors, various restaurants, manufacturing companies and private individuals, in all, about 100.

Both miners and operators are now engaged in attempting to hold war-gained markets which center on the home-heating field.

• **COMPETITION.**—While the union opens up its attack to ward off use of oil in the very heart of "coal country." The Anthracite Institute Bulletin, organ of the industry, shows how seriously it views the inroads made by oil burners. A recent issue noted that oil burner installations throughout the country for the first seven months of 1947 were 95% ahead of those in 1946, and estimated that 60% of the 397,392 units put in were in anthracite's primary market area.

Breaking down the installations percentage-wise, the Bulletin estimated that "approximately 157,000 solid fuel burning plants in our marketing area may

A-P SCORES AGAIN
IN
REFRIGERANT CONTROL ACCURACY



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Models 211 and 212 VALVES

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A necessity and money-saving convenience on many modern systems, the new A-P Model 211 and 212 Expansion Valves avoid all those service problems caused by motor overloading at sharp peak loads or during pull down. Amazingly simple in construction, Models 211 and 212 use only one spring and one needle pin. A unique and entirely new diaphragm unit limits the operation suction pressure to a predetermined setting. Should the pressure build up to a point approaching overload, this diaphragm actuates the needle pin, allowing the valve needle to move towards the closed position, until the overload condition has passed. Then the valve operates in the normal manner to control superheat through the thermostatic bulb and sensitive large-area diaphragm.

Model 211 is non-adjustable. Model 212 is adjustable, and may be used in place of any standard thermostatic expansion valve of similar capacity. Types are available to limit pressures to 15, 40, and 55 lbs. Freon 12. Capacities: ½, 1 and 1½ tons Freon.

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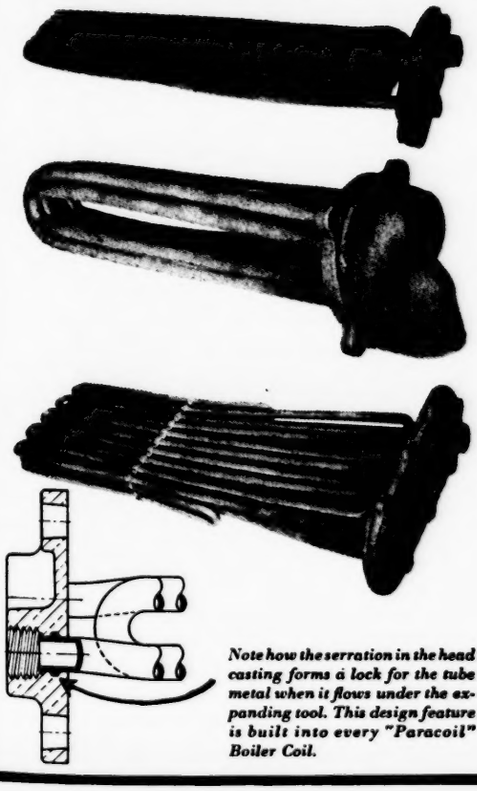
Submerged Boiler Coils
for HOME and INDUSTRY



Note how the serration in the head casting forms a lock for the tube metal when it flows under the expanding tool. This design feature is built into every "Paracoil" Boiler Coil.

"Paracoil" has been building dependable boiler coils since 1915 for both home and industry. Here are a few of the varied types of instantaneous tankless water coils we are now fabricating for domestic residence boilers.

All head castings have SERRATED TUBE CONNECTIONS—another "Paracoil" design feature! You are doubly protected against leaks with a "Paracoil" Boiler Coil!

"Paracoil" Boiler Coils are designed for use with or without a storage tank. The larger industrial models are built with integrally cast water chambers and tube sheets to any desired capacity.

"Paracoil" manufactures many types of heating and ventilating accessories—all distributed through franchised agents. Some lucrative territories are still available. If interested, write our Elizabeth office for details.

DAVIS ENGINEERING CORPORATION
1058 EAST GRAND STREET, ELIZABETH 4, NEW JERSEY
30 ROCKEFELLER PLAZA, NEW YORK 20, N. Y.
BUY OR SPECIFY "PARACOIL"

have been converted to oil in the first seven months of this year."

• **UNIT OUTPUT.**—Within the past month, officials of the United Mine Workers in the anthracite region have called upon members to cut absenteeism and increase their man-output to lower production costs and improve the competitive position of anthracite. A letter to members signed by John L. Lewis and other members of the union hierarchy pointed out that the prosperity of the miners is inexorably tied to the ability of their product to meet the unit cost, quality, and sales advantages of other fuels. Miners are reassured that their efforts will not serve merely to enrich the operators.

AIR CONDITIONING

of theaters discussed by Society of Motion Picture Engineers.

Data regarding the air conditioning and air disinfection of theaters were presented by four speakers at a ventilating and air conditioning session during the convention of the Society of Motion Picture Engineers held at the Hotel Pennsylvania, New York City, October 24.

• **AIR HANDLING.**—In discussing the air conditioning of motion picture theaters, Dwight B. Kimball, consulting engineer, said that basically air conditioning is but the outgrowth of ventilation. Carnegie Hall which was provided with a ventilating system, was easily converted recently to air conditioning. It has been proven that the air supply to theaters must be 24 cu ft per hr for the average design and 30 cu ft per hr for the deluxe theater. Air should be sprayed over the seating area. The balcony must be considered as three zones each having a separate return air. About 25% outside air is considered adequate even for odor removal.

Clean air is desirable for patrons as well as for decorations. While electrostatic filters are more desirable than the customary types, more space is required.

Air conditioning of projection rooms is governed by local regulations and those of the National Board of Fire Underwriters. A regulation of the NBFU states that the air conditioning system of such rooms shall not be connected with the system supplying other parts of the building.

Toilet rooms which are generally located off the lounges, draw air from the lounges and discharge the air to the outside.

• **SIZE OF SYSTEM.**—An empirical rule for the size of air conditioning systems is to allow 12 occupants per ton refrigeration for a 600-seat theater, 14 occupants per ton for 1,000 seats, 16 occupants per ton for 1,800 seats and 17 occupants per ton for the larger theaters. In no case does he advise to go over 17 seats per ton, regardless of the size of the theater. New York and other cities do not permit the use of direct expansion systems.

Air velocity should be limited to 1,200 fpm. Fan noise and maintenance were also considered. The noise level should not be above 35-40 decibles.

Mr. Kimball advised not to buy air conditioning systems on the basis of guarantees as it is difficult to obtain weather conditions to substantiate such guarantees.

• **GLYCOL.**—J. W. Seisselman of Air Purification Service described the use of glycol vapors as applied to theater air conditioning. Glycol mixtures were used in England in air raid shelters with some success. Before the United States entered the war, studies were made for the U. S. Public Health Service and it was found that not only was triethylene glycol the most effective agent in the mixture that was used in England, but that it was also the cheapest. He said that all the air in a building six stories high covering one city block could be sterilized by a pint of the chemical. The vapor is non-toxic, non-corrosive and has a great affinity for water. He said that glycol produces a feeling of freshness in the air and that it is believed that it destroys molds in ducts.

• **ULTRA VIOLET.**—Air disinfection in theaters by ultraviolet light was covered by Dr. L. J. Buttolph, General Electric Co. It is not practical to directly radiate in theaters because the light will irritate face and eyes. He said that at present a certain amount of blue light appears when the lamp is in operation so that a moonlight effect is produced in the darkness of the theater. However, if the reflectivity of the paint used is 25% or less, no problem exists. To those who wonder why it would not be possible to place the lamps in ducts, he pointed out that such a plan would only treat the make-up air.

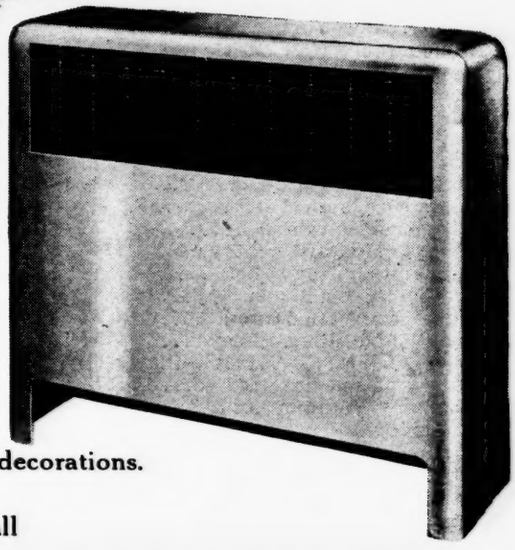
• **OPERATION AND MAINTENANCE.**—W. B. Cott, Westinghouse Electric Corp., discussed the economics of operation and maintenance of theater air conditioning systems. To explain why air conditioning systems cost so much today, he pointed out that using October 1939 as a base, labor rates had increased 4.8% in 1940, 54% in 1946 and 85.2% in 1947. Again using 1939 as a base, steel had increased 35% and copper 95%. Galvanized steel which formerly sold at 4 cents a pound is now priced at from 11 to 15 cents per pound.

Since air conditioning systems in theaters represent a sizable investment, it is important that the owners arrange for proper maintenance and service. Make sure that the operator has plenty of elbow room to service the equipment properly. Formerly, the cost of maintenance was 15 to 19 cents per seat per year; now it is from 23 to 27 cents. Because of the growing shortage of material and parts, it is important that equipment now installed be properly maintained and serviced.

In answer to questions from the audience directed to the speakers, it was pointed out that ultraviolet lights can be installed in recesses in the walls of modern theaters at an approximate cost of $1 per seat. In old theaters it is difficult to find a proper place for

(Continued on page 130)

Clean, Safe, Economical HEATING with AIRTHERM CONVECTORS



AIRTHERM CONVECTORS for hot water and steam heating systems provide the desirable combination of convection and radiant heating. The attractive cabinets can be readily finished after installation in any color to harmonize with interior decorations.

AIRTHERM CONVECTORS are available in three cabinet styles—Type F, free standing or partially recessed; Type W, wall cabinet; and Type S, sloping top wall cabinet—in a complete range of depths, heights, and lengths.

Have you recently investigated the advantages of specifying and installing convectors? Write for new bulletin describing AIRTHERM CONVECTORS.

AIRTHERM MANUFACTURING CO.

722 South Spring Ave. • St. Louis, Mo.

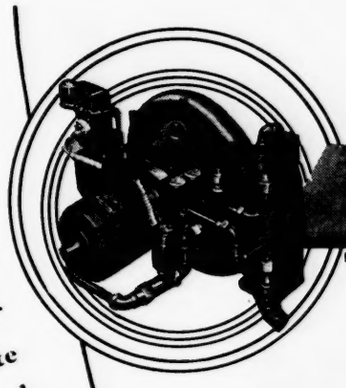
62

Cordial Invitation

OUR increased production facilities and personnel enable us to satisfy the requirements of a few additional competent organizations capable of selling, engineering, installing, and servicing our complete line of industrial horizontal rotary oil burners.

You are invited to investigate the unusual sales and merchandising advantages of the Ace Uniflow Oil Burner.

A profitable opportunity awaits the right organizations.



Ace UNIFLOW HORIZONTAL ROTARY

HORIZONTAL ROTARY

OIL BURNERS

FOR PERFECT HEAVY OIL BURNING UNDER ALL OPERATING CONDITIONS

Models Available

	GALLONS PER HOUR	BOILER H. P.	SQUARE FEET STEAM RAD.
MINIMUM	2	6	800
MAXIMUM	165	500	64,000

ACE ENGINEERING CO.

1435 W. 15th Street • Chicago, Illinois

Custom Engineered Oil Burning Systems Since 1931

DEGREE-DAYS FOR SEPTEMBER, 1947

HEATING AND VENTILATING's 21st Year of Publication of Monthly Degree-Day Data

City	September			Cumulative, Sept. 1, 1947 to Sept. 30, 1947			Heating Season[1]
	1947	1946	Normal	1947	1946	Normal	Normal
Abilene, Texas	0	5	0	0	5	0	2061
Albany, New York	185	91	72	185	91	72	6580
Albuquerque, New Mexico	0	12	0	0	12	0	4298
Alpena, Michigan	234	216	222	234	216	222	8299*
Anaconda, Montana	355	380	335	355	380	335	8357**
Asheville, North Carolina	80	35	0	80	35	0	4232
Atlanta, Georgia	23	14	0	23	14	0	2890
Atlantic City, New Jersey	87	19	0	87	19	0	5176
Augusta, Georgia	22	0	0	22	0	0	2161
Baker, Oregon	224	295	258	224	295	258	7163
Ba'timore, Maryland	87	11	0	87	11	0	4533
Billings, Montana	214	211	189	214	211	189	7119
Binghamton, New York	(a)	102	108	(a)	102	108	6808
Birmingham, Alabama	8	15	0	8	15	0	2352
Bismarck, North Dakota	275	263	222	275	263	222	9192
Block Island, Rhode Island	115	48	0	115	48	0	5788
Boise, Idaho	136	167	102	136	167	102	5552
Boston, Massachusetts	137	57	48	137	57	48	6045
Bozeman, Montana	327	309	336	327	309	336	8521**
Buffalo, New York	168	109	75	168	109	75	6822
Burlington, Vermont	206	129	144	206	129	144	7514
Butte, Montana	381	433	344	381	433	344	8235**
Cairo, Illinois	43	10	0	43	10	0	3909
Canton, New York	242	167	189	242	167	189	8020
Charles City, Iowa	171	170	93	171	170	93	7588
Charleston, South Carolina	4	0	0	4	0	0	1769
Charlotte, North Carolina	53	6	0	53	6	0	3120
Chattanooga, Tennessee	23	8	0	23	8	0	3118
Cheyenne, Wyoming	175	257	240	175	257	240	7466
Chicago, Illinois	107	56	76	107	56	76	6077
Cincinnati, Ohio	84	14	0	84	14	0	4684
Cleve'and, Ohio	127	62	27	127	62	27	6155
Columbia, Missouri	50	30	0	50	30	0	4922
Columbia, South Carolina	32	1	0	32	1	0	2364
Columbus, Ohio	112	35	0	112	35	0	5398
Concord, New Hampshire	236	142	168	236	142	168	7353
Concordia, Kansas	49	75	0	49	75	0	5315
Dallas, Texas	0	0	0	0	0	0	2256
Davenport, Iowa	105	73	0	105	73	0	6289
Dayton, Ohio	131	44	0	131	44	0	5264
Denver, Colorado	73	99	72	73	99	72	5874
Des Moines, Iowa	103	103	0	103	103	0	6384
Detroit, Michigan	150	85	42	150	85	42	6490
Devils Lake, North Dakota	339	315	276	339	315	276	9970
Dodge City, Kansas	19	64	0	19	64	0	5035
Dubuque, Iowa	142	114	36	142	114	36	6790
Duluth, Minnesota	336	304	270	336	304	270	9483
Eastport, Maine	239	199	276	239	199	276	8520**
Elkins, West Virginia	161	124	63	161	124	63	5697
El Paso, Texas	0	4	0	0	4	0	2428
Ely, Nevada	220	268	(a)	220	268	(a)	(a)
Erie, Pennsylvania	118	73	36	118	73	36	6273
Escanaba, Michigan	233	272	243	233	272	243	8771
Evansville, Indiana	81	34	0	81	34	0	4244
Fort Smith, Arkansas	5	6	0	5	6	0	3147
Fort Wayne, Indiana	154	87	0	154	87	0	5925
Fort Worth, Texas	0	0	0	0	0	0	2148
Fresno, California	2	0	0	2	0	0	2334
Galveston, Texas	0	0	0	0	0	0	1016
Grand Junction, Colorado	27	62	0	27	62	0	5548
Grand Rapids, Michigan	161	94	57	161	94	57	6535
Green Bay, Wisconsin	205	185	132	205	185	132	7825
Greensboro, North Carolina	72	20	0	72	20	0	3529
Greenville, South Carolina	42	10	0	42	10	0	3380
Harrisburg, Pennsylvania	125	47	0	125	47	0	5375
Hartford, Connecticut	158	52	48	158	52	48	6036
Hatteras, North Carolina	2	0	0	2	0	0	2571
Havre, Montana	286	279	270	286	279	270	8700
Helena, Montana	283	313	251	283	313	251	7894**
Houston, Texas	0	0	0	0	0	0	1157
Huron, South Dakota	182	193	123	182	193	123	8004
Indianapolis, Indiana	106	39	0	106	39	0	5298
Jackson, Mississippi	3	0	(a)	3	0	(a)	(a)
Kansas City, Missouri	41	37	0	41	37	0	4956
Kewanee, Illinois	171	48	78	171	48	78	6139
Knoxville, Tennessee	38	12	0	38	12	0	3670
La Crosse, Wisconsin	199	200	96	199	200	96	7322
Lander, Wyoming	180	266	276	180	266	276	7947

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.

Figures in this table, with nine exceptions, based on local weather bureau reports. Exceptions are Utica, Lewiston, Kewanee, Sioux City, and Jackson, figures for which are furnished through the courtesy of Coke Sales Department, Central New York Power Corp., Utica, N. Y.; Norman E. Ross, Bursar, Bates College, Lewiston, Me.; J. M. Hartman, Engineering Department, Kewanee Boiler Corp., Kewanee, Ill.; Alfred R. Wagstaff, Engineer, Sioux City Foundry and Boiler Co., Sioux City, Iowa, and H. M. Ludlow, Box 1368, Jackson, Miss., respectively; Anaconda, Bozeman, Butte and Livingston, Mont., through the courtesy of the Montana Power Company.

[Table concluded on Page 124]

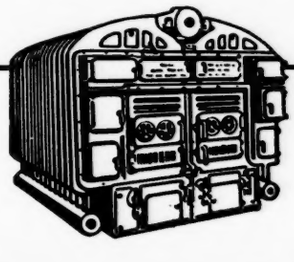
EXTRA EXTRA

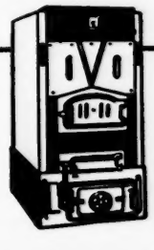
"Extra . . . Extra, Read all about it!"

H. B. Smith Cast-Iron Boilers are packed with *extra* heating surface . . . you can read all about it in H. B. Smith literature. The *extra* that means real fuel economy is in the many *more* square feet of *direct* fire surface packed into H. B. Smith boilers.

Match this heating surface with that of any cast-iron boiler of equal grate area and you'll see why H. B. Smith boilers extract more value from each fuel unit burned. Conventional cast-iron boilers have *single* water tubes on either side of the fire pot; large H. B. Smith boilers have *two* vertical tubes, leading into many other vertical and lateral tubes that are packed with water backed surface. That means all the flue gases get a chance to scrub along this *extra* heating surface, *more* heat units are transmitted to the water.

That, too, is why H. B. Smith boilers are outstandingly efficient for automatic firing. There is plenty of heating surface to efficiently absorb the heat generated by intense, blast-like oil, gas or stoker flames . . . even at peak firing rates. *Extra* H. B. Smith heating surface is your guarantee of economical boiler operation.





H.B. Smith

CAST-IRON BOILERS

THE H. B. SMITH CO., INC., 63 Main Street, Westfield, Mass. Offices and Representatives in Principal Cities

Degree-Days for September, 1947 (Concluded)

HEATING AND VENTILATING's 21st Year of Publication of Monthly Degree-Day Data

City	September			Cumulative, Sept. 1, 1947 to Sept. 30, 1947			Heating Season[1]
	1947	1946	Normal	1947	1946	Normal	Normal
Lansing, Michigan	199	128	114	199	128	114	7048
Lewiston, Maine	219	160	165	219	160	165	7707
Lincoln, Nebraska	84	67	0	84	67	0	5999
Little Rock, Arkansas	11	5	0	11	5	0	2811
Livingston, Montana	301	302	227	301	302	227	7245**
Los Angeles, California	1	0	0	1	0	0	1504
Louisville, Kentucky	61	19	0	61	19	0	4180
Lynchburg, Virginia	99	32	0	99	32	0	3980
Macon, Georgia	19	7	0	19	7	0	2201
Madison, Wisconsin	177	132	87	177	132	87	7429
Marquette, Michigan	270	261	225	270	261	225	8693*
Memphis, Tennessee	14	5	0	14	5	0	2950
Meridian, Mississippi	9	2	0	9	2	0	2160
Milwaukee, Wisconsin	179	140	84	179	140	84	7245
Minneapolis, Minnesota	198	195	93	198	195	93	7850
Modena, Utah	89	138	156	89	138	156	6562
Montgomery, Alabama	4	2	0	4	2	0	1884
Nantucket, Massachusetts	139	75	63	139	75	63	5957
Nashville, Tennessee	27	7	0	27	7	0	3507
New Haven, Connecticut	142	46	39	142	46	39	5895
New Orleans, Louisiana	0	0	0	0	0	0	1024
New York, N. Y.	105	19	50	105	19	50	5274***
Nome, Alaska	Degree-Days for June, 1947 (not previously published) are 434. Total for season (Sept. 1 to June 30, 1947) is 13513.						
Norfolk, Virginia	30	1	0	30	1	0	3350
North Head, Washington	220	234	255	220	234	255	5452**
North Platte, Nebraska	113	103	42	113	103	42	6366
Oakland, California	78	74	90	78	74	90	3143**
Oklahoma City, Oklahoma	0	16	0	0	16	0	3613
Omaha, Nebraska	88	84	0	88	84	0	6131
Oswego, New York	167	112	114	167	112	114	7088
Parkersburg, West Virginia	110	21	0	110	21	0	4775
Peoria, Illinois	109	76	3	109	76	3	6109
Philadelphia, Pennsylvania	95	12	36	95	12	36	4737***
Phoenix, Arizona	0	0	0	0	0	0	1405
Pittsburgh, Pennsylvania	131	32	0	131	32	0	5235
Pocatello, Idaho	181	215	156	181	215	156	6655
Portland, Maine	223	146	162	223	146	162	7218
Portland, Oregon	67	116	105	67	116	105	4469
Providence, Rhode Island	133	50	63	133	50	63	6015
Pueblo, Colorado	49	83	6	49	83	6	5514
Raleigh, North Carolina	57	5	0	57	5	0	3234
Rapid City, South Dakota	164	212	144	164	212	144	7118
Reading, Pennsylvania	119	29	0	119	29	0	5389
Red Bluff, California	6	1	(a)	6	1	(a)	(a)
Reno, Nevada	136	170	144	136	170	144	5892
Richmond, Virginia	77	11	0	77	11	0	3695
Rochester, New York	173	110	72	173	110	72	6732
Roseburg, Oregon	97	118	111	97	118	111	4428
Roswell, New Mexico	3	21	0	3	21	0	3484
Sacramento, California	8	2	0	8	2	0	2653
St. Joseph, Missouri	55	44	0	55	44	0	5161
St. Louis, Missouri	56	24	0	56	24	0	4585
Salt Lake City, Utah	110	137	18	110	137	18	5555
San Antonio, Texas	0	0	0	0	0	0	1202
San Diego, California	0	0	0	0	0	0	1645
Sandusky, Ohio	114	44	0	114	44	0	6208
San Francisco, California	136	117	114	136	117	114	3264**
Sault Ste. Marie, Michigan	310	286	276	310	286	276	9285**
Savannah, Georgia	5	0	0	5	0	0	1490
Scranton, Pennsylvania	162	68	60	162	68	60	6129
Seattle, Washington	127	141	186	127	141	186	4934**
Sheridan, Wyoming	200	222	264	200	222	264	8008
Shreveport, Louisiana	2	1	0	2	1	0	1938
Sioux City, Iowa	142	128	33	142	128	33	6898
Spokane, Washington	200	208	192	200	208	192	6355
Springfield, Illinois	80	40	0	80	40	0	5373
Springfield, Missouri	44	41	0	44	41	0	4428
Syracuse, New York	182	123	96	182	123	96	6893
Tacoma, Washington	166	192	207	166	192	207	5181**
Terre Haute, Indiana	114	39	0	114	39	0	4872
Toledo, Ohio	156	79	9	156	79	9	6077
Topeka, Kansas	39	49	0	39	49	0	4969
Trenton, New Jersey	120	24	0	120	24	0	4933
Utica, New York	172	115	182	172	115	182	6796
Valentine, Nebraska	140	156	93	140	156	93	7039
Walla Walla, Washington	68	103	30	68	103	30	4808
Washington, D. C.	87	11	0	87	11	0	4626
Wichita, Kansas	17	36	0	17	36	0	4673
Williston, North Dakota	316	292	270	316	292	270	9323
Winnemucca, Nevada	151	173	192	151	173	192	6427**
Yakima, Washington	116	168	120	116	168	120	5599

(a) Data not available.

*Includes August.
**Includes July and August.
***New 48-year normal covering 1898 to 1946.

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.

Customers PROFIT in *Comfort* and *Economy*

YOU PROFIT in *Cash* and *Goodwill*

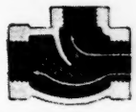
. . . by converting to
HOT WATER HEATING AT ITS BEST!

This is an era of conversion . . . and it's easy to convince many customers that they will reap big profits in comfort and operating economy by converting antique heating systems to *hot water at its best!* That means Triplex Heating . . . for the advanced design and precision construction of all Triplex Flow Control Units assure the most efficient, dependable operation possible. Hot water is automatically circulated, abundant heat supplied to radiators instantly . . . whenever and wherever needed. And any Triplex installation provides a constant supply of domestic hot water for kitchen, laundry, bath and lavatory both summer and winter.

Triplex installations reap profits for you, too . . . substantial cash profits on the jobs . . . valuable profits in goodwill that only satisfied customers can bring. Yes, *Triplex means hot water at its best* . . . for everyone concerned. Push it for conversions as well as new installations. You'll win both ways. Write for latest catalog today.

Direct Drive Circulators

Flow Control Valves



One-Pipe Distributors

Relief and Pressure Reducing Valves

Indirect Heaters

Triplex

HEATING SPECIALTY CO., INC.
PERU, INDIANA

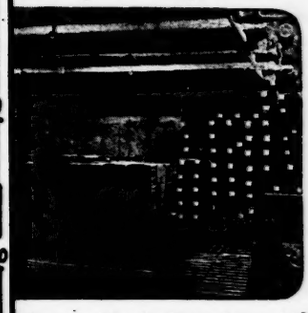
In addition to units illustrated, the Triplex line includes—Direct Drive Horizontal Circulators, Indirect Drive Circulators, Expansion Tanks, Air Eliminators, Water Mixing Valves, Air Vent Valves, Tankless Heaters, Control Units, Electrical Controls, Barometric Draft Dampers, Sump Pumps.

SEE YOUR WHOLESALER TODAY

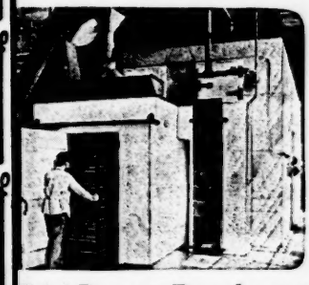
Pratt & Whitney Keep This Room at 68 deg. F, the Year 'Round.

Hold Any Temperature You Want with

One-man One-shift Semi-Automatic Plant Makes 42 Tons of Ice Daily.



Frozen Foods Stored at -10° F by Birds Eye-Snider.

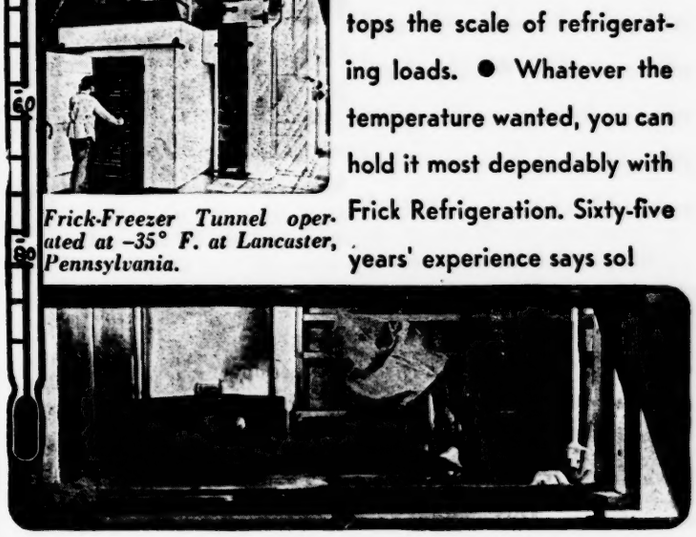


Frick-Freezer Tunnel operated at -35° F. at Lancaster, Pennsylvania.

TRADE FRICK MARK

Refrigeration

"Cold" down to 130 degrees below zero F. is now common in research and test work. Penicillin is dried at minus 75. Foods are quick-frozen at minus 30 to minus 60: are stored at zero to minus 20. Ice is frozen commercially in brine at 16. Fresh foods are held at 34 to 36. Drinking water is cooled to 45. Air conditioning, at 70 to 85, tops the scale of refrigerating loads. ● Whatever the temperature wanted, you can hold it most dependably with Frick Refrigeration. Sixty-five years' experience says so!



Test at -76° F. in a Refrigerated Laboratory of Bendix Radio.

DEPENDABLE REFRIGERATION SINCE 1882

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NEW CATALOGS

Unit Heaters

Kennard-engineered unit heaters, horizontal type, are described in bulletin No. 4710 which includes tables of steam capacities and specifications, dimensions, hot water capacities, and other selection data. Drawings illustrate piping connections for installations in low-pressure vapor, high-pressure steam, low-pressure gravity, and two-pipe wet return systems. Other Kennard products, including expansion coils and blower units, are also illustrated.—*Kennard Corporation, 1819 So. Hanley Rd., St. Louis 17, Mo.* 386

Adjustable Air Diffusers

A new catalog and engineering data book on Kno-Draft adjustable ceiling air diffusers and accessories contains all the information and data necessary for the proper selection of air diffusers. This handbook has been compiled for architects, engineers and contractors and includes much information of a general nature on the subject of air diffusion. The advantages of Kno-Draft diffusers are fully discussed.—*W. B. Connor Engineering Corp., 114 East 32nd St., New York 16, N. Y.* 387

Fan-Type Heater

The new Emerson-Electric combination radiant and fan-type heater is pictured and described in a 4-page bulletin, No. X5838. Included are specifications and construction features and suggestions for uses of this new heater.— *The Emerson-Electric Company, St. Louis, Missouri.* 388

Packings

A new catalog section describes the Goodrich lines of packings. The section covers various types of compressed asbestos sheet packing, rubber sheet packing, cloth inserted packings and Koroseal sheet packing. —*The B. F. Goodrich Co., Akron, Ohio.* 389

Steam Generator

Bulletin 28, eight pages, three colors, explains the principle of cyclonic combustion and heat transfer employed in the Cyclotherm line of steam generators. —*Cyclotherm Corporation, 90 Broad Street, New York 4, N. Y.* 390

Oil Burner Valves

Oil burner valves and fittings for copper tube connections to oil burners are described and illustrated in a 12-page, two-color bulletin, Folder 102—*Hays Mfg. Co., Erie, Pa.* 391

Instruments

Specialized testing and measuring equipment is described in a 44-page bulletin. Items include time, speed, torque, force, strain, and thickness measuring equipment, and instruments for insulation testing, balancing, pressure measuring, and circuit testing.—*General Electric Co., Schenectady 5, N. Y.* 392

V-Belt Sheaves

Sizing and capacity tables for cast iron and standard sheaves for V-belts are included in a 16-page, two-color bulletin. Application drawings are shown and prices are listed.—*American Gear & Mfg. Co., Chicago, Ill.*.....393

Torque Indicators

Six torque indicator models are listed in a new bulletin, 46T. These indicators are described as tension wrenches which are, in effect, socket wrench handles which measure the amount of pull exerted by the workman.—*Blackhawk Manufacturing Company, Milwaukee 1, Wis.* 394

Ram-Feed Stokers

An 8-page bulletin illustrates and describes Types R and RI Ram-Feed stokers. Capacities with single retorts are 150 to 560 boiler hp. Twin retorts are available for larger requirements. May be equipped with automatic air volume and coal feed controls. Photographs, cut-away drawings and installation layouts supplement descriptions of mechanical features.—*The Brownell Company, 410 North Findlay St., Dayton 1, Ohio.* 395

Pneumatic Valves

"Pneumatic Control Valves and Controller Accessories," a 36-page Bulletin 277-2 has been issued in a new edition, with all contents brought up to date. A color page shows the various identifying enamel finishes offered on Stabilflo valves. Plates and tables give specifications for control valves, needle type valves, poppet valves and butterfly valves. Separate sections describe the Vernier Valvactor for high-accuracy positioning of valve plungers.—*The Foxboro Co., Foxboro, Mass.* 396

Nickel and Nickel Alloy Tubing

Catalog Section 10 on nickel and nickel alloy small tubing (.010 in. to ⅝ in. max. O.D.) gives comparative tabular data regarding physical and chemical characteristics of nickel, monel, "K" monel, inconel and cupro-nickel. Working tables are included on commercial tolerances, standard production limits, ordering data, and Superior tempers.—*Superior Tube Co., 2138 Germantown Ave., Norristown, Pa.* 397

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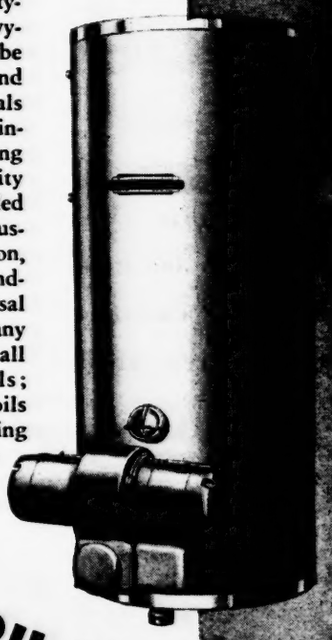

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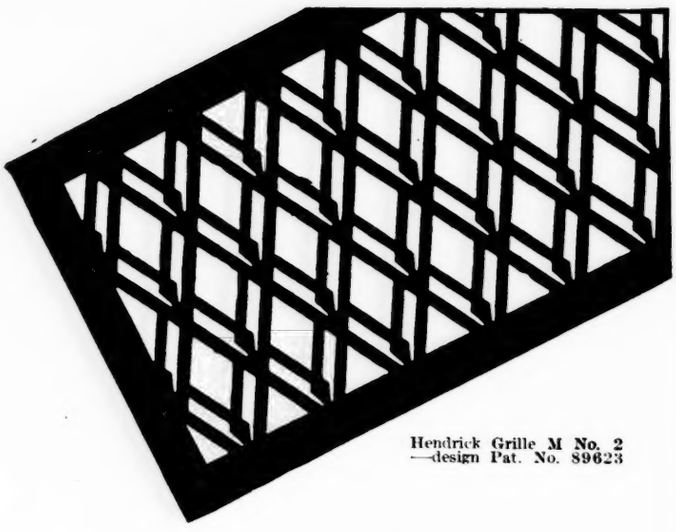

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Steam Specialties

A four-page bulletin on steam traps, steam separators and other steam specialties which are immediately available describes three types of Cub "Airxpel" steam traps suitable for pressures from 0 to 250 lb per sq in. discharging at varying rates from 300 to more than 6000 lb of water per hour on intermittent service or double these rates on continuous discharge. —*Wright-Austin Co., 315 West Woodbridge St., Detroit 26, Mich.* 398

Steam Generators

A 12-page bulletin illustrates and describes the Springfield line of 2-drum bent tube and sectional header cross drum boilers. Designs featured include Springfield Type M standardized boilers for small and medium size plants, with data and dimensions.—*Springfield Boiler Co., 1945 East Capitol Ave., Springfield, Ill.* 399

Heating Specialties

An eight-page bulletin describes Trane convectors, unit heaters, valves, hot water pumps, and fans for a wide variety of heating applications.—*The Trane Co., LaCrosse, Wis.* 400

Fittings

Bulletin 48, tab indexed, 62 pages, loose-leaf bound, describes the Gedney line of fittings for wiring, and other electric services. Classifications include conduit fittings, box connectors, entrance fittings, ground fittings, etc.—*Gedney Electric Co., RKO Building, Radio City, New York 20, N. Y.*.....401

Stainless Steel Fabrication

The fabrication of Enduro stainless steels is covered in a 50-page booklet which describes techniques for handling sheet steel. Brazing, soldering, welding, grinding, polishing, and riveting are discussed with tables and engineering data.—*Republic Steel Corp., Chrysler Bldg., New York 17, N. Y.* 402

Power Boilers

Operation and care of Kewanee power boilers are described in a four-color booklet. High-pressure boilers only are discussed with respect to construction, identification, putting in service, firing, care of grates, etc.—*Kewanee Boiler Corp., Kewanee, Ill.* 403

Rivnut Data Book

The fourth edition of the Rivnut Data Book covering types, sizes, selection, installation techniques, applications, and strength characteristics of the B. F. Goodrich Rivnut is now available.—*B. F. Goodrich Co., Akron, Ohio.* 404

Tube Bending

The Pines principle of tube bending in the fabrication of pipe coils, tubular furniture, and various shapes of metal parts, is described in an 18-page catalog illustrating a line of bending machines. Capacity charts and procedure sketches are included.—*Pines Engineering Co., Inc., Walnut St., Aurora, Ill.*.....**405**

Steam Purifiers

Publication 2725 is a new edition of a 12-page bulletin describing Cochrane steam purification units for the removal of moisture and solids in steam lines. Application data, hook-up diagrams, and installation photographs are included.—*Cochrane Corp., 17th St. & Allegheny Ave., Philadelphia 32, Pa.* **406**

Dust and Fume Collectors

The Centri-Merge line of swirl type dust and fume eliminators is described in a 12-page folder. Sectional drawings, photographs, and layout blueprints illustrate collectors with capacities up to 30,000 cfm.—*Schmieg Industries Inc., 300 Piquette Avenue, Detroit 2, Michigan.* **407**

Piping Engineering

A bulletin of Piping Engineering, noted as fifth in a series, covers welded joints with respect to material preparation, pertinent code regulations, alignment and assembly techniques.—*Tube Turns Incorporated, Louisville 1, Ky.* **408**

Protective Coating

A 12-page catalog gives details of the chemical and physical properties, application instructions and other data on protective coatings.—*Amercoat Div., American Pipe and Construction Co., P. O. Box 3428, Terminal Annex, Los Angeles 54, Calif.* .. **409**

NOVEMBER, 1947

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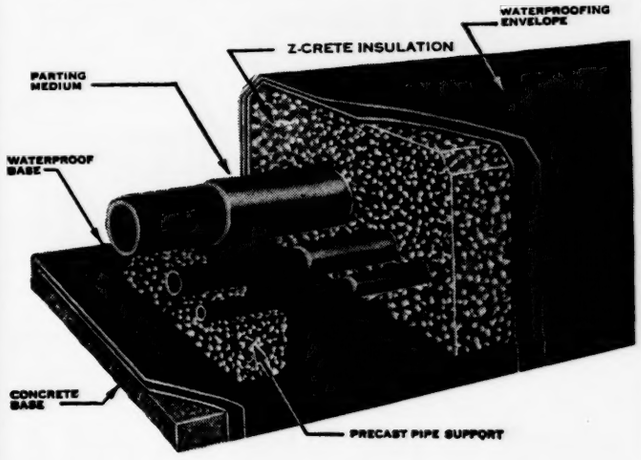

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Rapid operation . . . Hairline Accuracy . . . the use of Kling Rotary Shears marks the latest development in cutting mild steel, and sheet metal, up to 1-inch with amazing savings in time, labor, and production costs.

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AIR CONDITIONING THEATERS

(Continued from page 120)

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Kennard Corporation, 1819 South Hanley Road, St. Louis 17, Mo., manufacturers of heating, air conditioning, and refrigeration heat transfer equipment have recently appointed several manufacturer's representatives. **David S. Falk,** 3049 East Grand Blvd., Detroit 2, Michigan, now represents Kennard in the State of Michigan. During the last eleven years he has been associated with the heating and air conditioning industry in the Detroit area. A sales agreement has been made with **M. A. Schneider,** president of The Consolidated Engineering Sales Co., operating in the general Cincinnati trade area with main office at the Southern Bank Building, Cincinnati 2, Ohio and with two branch offices for territorial coverage. For the general Texas area, Kennard announces a sales agreement with the **Rodgers-Barbeck Co.,** 2021 Cedar Springs, Dallas, and 2019 Congress Avenue, Houston, Texas.

L. H. Matthes has been appointed manager of marketing for **General Electric's** Air Conditioning Department with headquarters at Bloomfield, N. J. Mr. Matthes, who will be responsible to the vice-president directly, will supervise all advertising and sales promotion, marketing services, and commercial engineering and sales for the G. E. department. Graduating from the G. E. test engineering course in 1922, Mr. Matthes spent two years as design engineer before becoming a sales engineer in the company's West Virginia office. In 1928 he was made manager of the Apparatus Department office in Beaumont, Texas. Prior to his present appointment Mr. Matthes was district manager for apparatus products in the Texas area.

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Company.....

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Wagner Electric

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No. 120A

Series No. 100

GRAVITY REGISTER

with Removable Face for Gravity or Air-Conditioning Installations

Series 100 registers, with removable face, have modern streamlined styling, large free area and minimum resistance. To assure maximum free area, the horizontal fins are set at the factory. The fins are flexible and may be readily adjusted up or down to any desired angle. Available in all sizes required for all new or old gravity installations or as a replacement register when changing an existing gravity system to air conditioning.

We also carry a complete line of Air-Conditioning Registers and Grilles

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CHELSEA FANS & BLOWERS

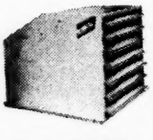
IND–INDUSTRIAL
24" to 60"
5700 to 32000 CFM

DXB—BOOSTER
16" to 36"
2000 to 9500 CFM

OPJ-OCTOPUS JR.
3-4" Hoses
CFM Per Hose
10' = 412 50' = 297

AA—UTILITY
20" & 24"
3400 to 4800 CFM

CP— BLOWER
3 Sizes
650 to 1020 CFM



PH—PENTHOUSE
24" to 48"
5700 to 21500 CFM

—for every ventilating requirement in

FACTORIES, SHOPS, OFFICES, SCHOOLS, INSTITUTIONS, etc.

REMOVE fumes, odors, steam and excessive heat.
PREVENT waste and plant shutdown on those hot, humid summer days.
INSURE maximum efficiency and working comfort for plant personnel.
PLAN NOW your ventilating and cooling installation.

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INSIST ON CERTIFIED RATINGS! Air delivery ratings of all CHELSEA products are determined by the test method prescribed by the PROPELLER FAN MANUFACTURERS ASSOCIATION and the A. S. of H. & V. E.

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CHELSEA

Crocker-Wheeler Electric Mfg. Co. of Ampere, N. J., a division of Joshua Hendy Corporation, has announced the appointment of **Paul J. Moore** to the newly created position of director of sales and engineering. A graduate electrical engineer, Mr. Moore has been engaged in the engineering and selling of electric motors and generators for 24 years. Mr. Moore was formerly connected with the Motor Engineering and Sales Departments of General Electric Co., Schenectady, N. Y. He was subsequently sales manager of Imperial Electric Co., Akron, Ohio; manager of sales and engineering of Star Electric Motor Co., Bloomfield, N. J., and manager of the Commercial Department of Watson-Flagg Machine Co., Paterson, N. J. Mr. Moore attended both Massachusetts Institute of Technology and Syracuse University, graduating from Syracuse in 1923. This was followed by graduate work at Union College in Schenectady, N. Y.

P. J. Moore

The James F. Lincoln Arc Welding Foundation has announced the appointment of **Professor R. S. Green** of the Department of Industrial Engineering of Ohio State University as editor-in-chief of the foundation's new book "Design for Welding," a survey of developments in welded design based on outstanding papers submitted in the foundation's $200,000 Design-For-Progress award program which closed June, 1947, the results of which will be announced in the near future.

Georges Faurie has been appointed general sales manager for the Heating and Appliance Division, **Evans Products Co.**, Plymouth, Mich. For the last three years Mr. Faurie has served as advertising, merchandising and sales promotion manager of the Home Radio Division, Westinghouse Electric Corp., in Sunbury, Pa.

John E. Payne, formerly manager of industrial sales, has been named manager of all industry sales departments for the **Westinghouse Electric Corp.**, with general over-all responsibility for sales of equipment to all industries. **R. S. Kersh**, who has been manager of the company's Houston, Texas, office since 1942, was named manager of industrial sales to succeed Mr. Payne. Both men will be located at the Westinghouse East Pittsburgh, Pa., plant.

Sporlan Valve Co. will be located in the new **Sporlan** Building, 7525 Sussex Avenue, St. Louis 17, Missouri. This change of address is effective as of October 6, 1947. The new Sporlan Building will contain the executive and general offices and the shipping and repair departments.

Jules P. Soudain of New Orleans and **James E. Nighbert** of Knoxville, Tenn., have joined the Southeastern regional appliance sales staff of **Rheem Manufacturing Co. Edward J. Hutchings** of Boston has been named New England district manager of appliance sales for **Rheem**, and **Vaughn G. Gooding** has joined the company as sales representative in the New England territory.

Stratton & Terstegge Co., Inc. announces the purchase for the **Anchor Division** of the Round Oak trade name, patterns, inventory and supplies for gas and electric ranges from the Dowagiac Foundry Division of Kaiser-Frazer Corp.

Leo M. Flanary, Modine Mfg. Co., Chattanooga, Tenn., representative since March 1, 1946, has now been appointed to also represent Modine in the company's Nashville territory. He will continue to make his headquarters at 938 Volunteer Bldg., Chattanooga.

Six new appointments in the commercial organization of **General Electric's** refrigeration machine and remote equipment divisions have been announced at the Air Conditioning Department headquarters in Bloomfield. **W. G. Cox** has been appointed assistant to the sales manager; **J. L. Roth** has been placed in charge of commercial activities at the Ft. Wayne, Ind., plant; and **W. F. Benoit** has been named sales manager, manufacturer sales. **R. M. Bukaty, H. R. Walker**, and **Milton Kalish** have been appointed product specialists.

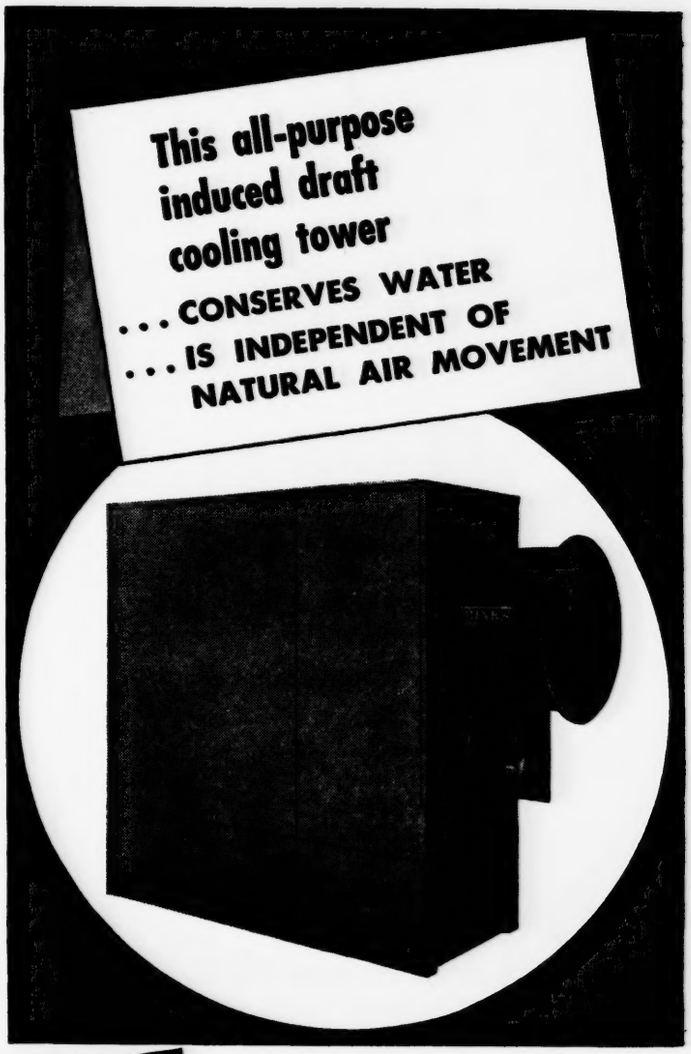
David S. Miller of Lancaster, Pa., was elected president of the **Producers' Council**, national organization of building products manufacturers, at the Council's recent annual meeting in New York City. Mr. Miller succeeds Tyler S. Rogers of Toledo, Ohio. Mr. Miller is sales promotion manager of the Building Materials Division of the Armstrong Cork Co. and has been associated with that company since 1937 when he graduated from Ohio State University. He was born and received his education in Columbus, Ohio. **F. L. Riggin, Jr.** of the Mueller Brass Co., Port Huron, Mich., and **C. R. Raquet**, of the Detroit Steel Products Corp., Detroit, Mich., were named as vice-presidents of the association. **Charles M. Mortensen** of Los Angeles has been appointed executive secretary of the Council.

The **American Welding Society** has elected **Harold O. Hill** as its president for the year 1947-48. Mr. Hill assumed that office on October 24 at the conclusion of the 1947 annual meeting at the Hotel Sherman, Chicago. Mr. Hill is assistant chief engineer, Fabricated Steel Construction, of the Bethlehem Steel Company, at Bethlehem, Pa. Mr. Hill has held many offices in the society and has been on numerous important committees during the 16 years he has been a member. He is also a member of the A.P.I., A.S.T.M., A.I.E.E., A.S.C.E., A.S.M.E., A.W.W.A., a registered professional engineer in the State of Pennsylvania and past-president of the Bethlehem Rotary Club.

The election of **E. G. Bailey**, vice-president of The Babcock and Wilcox Company, New York, as the next national president of **The American Society of Mechanical Engineers**, was announced at the society's headquarters.

Announcement is made of the appointment of **James D. Lightbody, Jr.**, as assistant to President Lyle C. Harvey of **Bryant Heater Co., Cleveland, Ohio.** A Magna Cum Laude graduate of Harvard in 1940, Mr. Lightbody began his career as design and factory engineer at United Shoe Machinery Co., Beverly, Mass. In 1941 he joined Thompson Products Co., Cleveland, and by 1944 he became a general foreman, after which he entered the army and served as a maintenance instruction officer, Ordnance, at Aberdeen Proving Ground, Maryland. He returned to Thompson Products in 1946, where he remained until recently as divisional and staff industrial engineer and assistant to the factory manager of the valve division.

J. D. Lightbody, Jr.



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34 STANDARD SIZES

Binks Type 2-D Horizontal Induced Draft Cooling Towers are ideal units for use either indoors or outdoors. They create their own draft, making them efficient in tight spots where the natural movement of the air is restricted. The ratio of air to water and travel flow is controlled to provide maximum cooling at the lowest possible cost. Single fan sizes are available with capacities from 3 to 150 G.P.M. Twin fan sizes have capacities from 180 to 300 G.P.M.

There's a Binks Cooling Tower for every water cooling job

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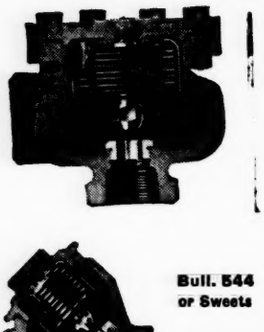
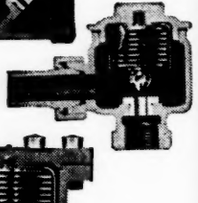
DOES AIR SLOW UP YOUR STEAM LINES?

Nicholson 'Fast Action' Traps Discharge Air as Efficiently as Water



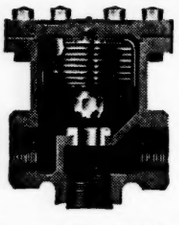
The positive, frequent intermittent action of Nicholson steam traps tends to pull air out of the system to the trap, where it is readily eliminated, speeding heat transfer. Installations result in production increases up to 30%.

Bull. 544 or Sweets



5 Types for Every Application

Extreme simplicity makes unit trapping economical. 5 types for process, heat, power; size ¼" to 2"; pressures to 225 lbs. Bull. 544.



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Valves ★ Traps ★ Steam Specialties

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PIONEERS IN THE MANUFACTURE OF COMPLETELY PACKAGED AIR-CONDITIONERS*
Manufacturers of:
• Evaporative Condensers
• Air Handling Units for Heating and Cooling
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EXCELLENT DELIVERY
Recent increases in our production facilities enable us to make rapid delivery on most of our products.
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513 N. BLACKWELDER OKLAHOMA CITY, OKLA.
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* with evaporative condensers, U. S. Pat. No. 2,297,928.

Appointment of **Thomas F. Gessner** as advertising manager of the **United States Radiator Corp.** and its **Pacific Steel Boiler Division** has been announced. Mr. Gessner came to U. S. Radiator from the Goodspeed-Detroit Co. where he was sales manager. Previously, he had been assistant director of public relations at Evans Products Co., and, for 24 years, had been associated with the Detroit Edison Co., first as editor of employee magazine and for 17 years as advertising manager.

Marlo Coil Co. of St. Louis announce the appointment of **W. J. Bagley** as their representative for St. Louis and surrounding territory. He was formerly Midwestern representative for Superior Valve and Fittings Co. of Pittsburgh and during the war an engineer with the Army Air Forces Aeronautical Chart Service.

Paul S. Park has been named manager of the Engineering Service Department of A. M. Byers Co., Pittsburgh, manufacturers of wrought iron. A graduate of the University of Pittsburgh in mechanical engineering, Mr. Park served as an officer in the Army Air Forces during World War II. He has been with A. M. Byers nearly four years. Besides handling engineering problems dealing with the use of wrought iron pipe and plate in industrial, marine, railroad, refrigeration, construction, petroleum, sewage and water-works fields, Mr. Park's department has been instrumental in the development of radiant heating and snow-melting systems.

P. S. Park

The appointment of two new representatives has been announced by the **Automatic Devices Co.**, makers of The Weather-Man, outside type temperature control. **John E. Brecht & Co.**, 2107 Central Avenue, Cincinnati 14, Ohio have been assigned the territory including Southwestern Ohio, Central and Eastern Kentucky. C. W. Gates Associates, Baldwin, Kansas, will represent the company in Kansas and Missouri.

Joseph H. Shoemaker, formerly in the sales department of the **Williams Oil-O-Matic Division of Eureka Williams Corp.**, Bloomington, Ill., has been appointed a district representative in the New England sales division.

Austin F. McCormack, of the New York office of **The Permutit Co.**, has been named sales engineer in charge of the water conditioning firm's newly established Rochester office in the Commerce Building, 119 Main Street, East, Rochester.

R. G. Vanderweil has become associated with **Slocum & Fuller**, consulting engineers, 21 East 40th Street, New York 16, N. Y. For the past few years, Mr. Vanderweil has been with the Chase Brass & Copper Co. where, as project engineer, he handled the development work in connection with that company's radiant heating research program. Previously, Mr. Vanderweil had been heating and power plant engineer with the Austin Co., Combustion Engineering Corp. and Baker Smith & Co. in New York City. Mr. Vanderweil holds a master's degree in mechanical engineering of the University of Vienna (he spent the first seven years of his practical experience in Austria). He is a member of ASHVE and ASME, and holds a professional engineer's license in the State of Connecticut.

Approximately 20,000 men, women and children attended a community-wide open house held at the huge new plant recently acquired from the War Assets Administration by **Carrier Corp.** as a part of its $12 million expansion program in the manufacture of air conditioning, refrigeration and industrial heating equipment. The principal building on the property covers more than 13 acres under a single roof. Built by the U. S. Navy in 1942 for the production of turbines and later used for jet propulsion equipment, the plant acquired on October 1, is already in partial operation, including three 800-foot-long assembly lines for the manufacture of centrifugal compressors. The new plant will give Carrier 600,000 sq ft of manufacturing space in addition to the 1,000,000 sq ft in its older factory in Syracuse.

Edwin L. Hall, coordinator of gas production research and secretary of the general research and planning committee of the **American Gas Association** has been appointed director of the AGA Testing Laboratories at Cleveland and Los Angeles as of November 1st. Mr. Hall will succeed R. M. Connor who is resigning from active duty as director because of ill health. Mr. Conner will be retained in the capacity of laboratories consultant. Mr. Hall joined the American Gas Association in May, 1945 as Coordinator of Gas Production Research.

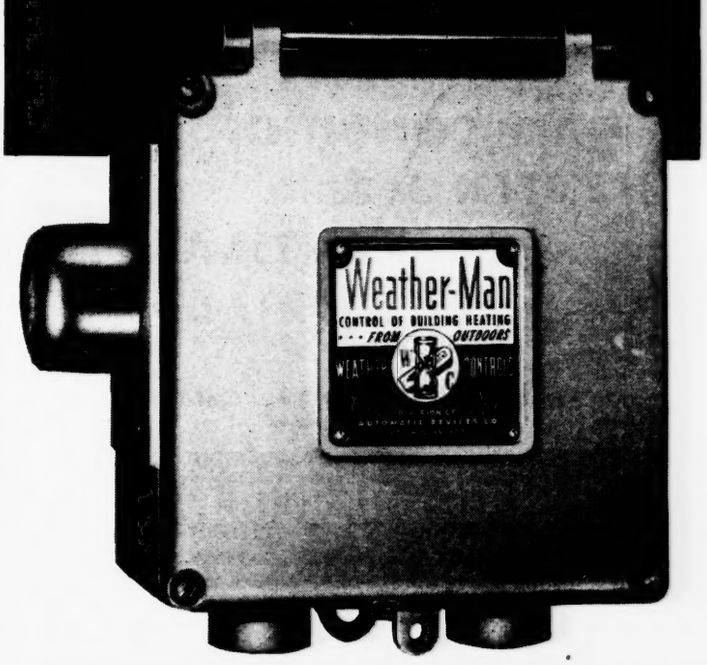
M. W. Crew, manager of the industrial control division of **Perfex Corp.,** has been elected to the office of first vice president of the **Smoke Prevention Association of America.**

Organization of a fifth region effective November 1 for the field organization of the **Allis-Chalmers** general machinery division under **Arch J. Cooper,** New York district office manager, has been announced. Formation of the first four regions to be established under the company's long-range sales plan was revealed earlier this year.

The Burkay Co., Toledo, Ohio announces the appointment of **Allison F. H. Scott** as vice-president in charge of sales of their heating products division. Mr. Scott's early association in the heating business has been with the American Radiator Co., New York, as assistant eastern sales manager, Anthracite Industries, Inc., New York, as assistant to president, and Hoffman Specialty Co., Waterbury, Connecticut, New York and Indianapolis, as vice-president and general sales manager. From 1934 to 1937 Mr. Scott was loaned by the American Radiator Co. to the Federal Housing Administration, Washington, as special assistant to the deputy administrator. In the fall of 1941 Mr. Scott was loaned by Hoffman Specialty Co. to become principal consultant to O.P.A. in Washington during the early formation days prior to Pearl Harbor. Upon completion of over-seas service as an Army Colonel in anticraft artillery, Mr. Scott returned to the Hoffman Specialty Co.

Harold H. Reed formerly assistant manager of **Air Reduction's** Metropolitan (New York) District has been appointed manager of this district. Mr. Reed succeeds W. S. Schoenthaler, who is retiring after thirty-seven years of continuous service with the company. Mr. Reed entered the employ of the Air Reduction Sales Co. in 1923 as a member of the gas sales department. **Oren M. Donohue** formerly assistant sales manager has been appointed assistant manager, succeeding Mr. Reed. Mr. Donohue joined the company in 1925 as a member of the Credit Department. **William B. Brower** who joined the company in 1922 now succeeds Mr. Donohue as assistant sales manager.

A NEW SIMPLIFIED WAY FROM THE OUTSIDE!



...AT NO MORE THAN THE USUAL COST OF ORDINARY THERMOSTATIC CONTROL

Here, at last, is a *simplified* "outdoor type" temperature control with startling new features. It is ideal for apartments, commercial and industrial buildings, yet so low in cost that even small homes can enjoy the comfort and operating economy possible only with an "outdoor" temperature control.

The Weather-Man is amazingly easy to install. No need to cut through floors, rip out plaster—wiring is all in the basement. Where else can you find a single, simple unit which *automatically* performs all these control functions—

1. **Starts up the heating plant in the morning—earlier on cold days, later on mild days.**
2. **Has the building warm at whatever time is desired.**
3. **Operates the heating plant just often enough during the day to keep building temperature constant.**
4. **Shuts off the heating plant at night—earlier on mild days, later on cold days.**
5. **Stops all heating completely when outdoor temperature rises above 65°.**

DELIVERY NOW!
The Weather-Man is available for immediate delivery through automatic heating equipment wholesalers and distributors.
TERRITORIES OPEN—WRITE!

Send today for descriptive literature.

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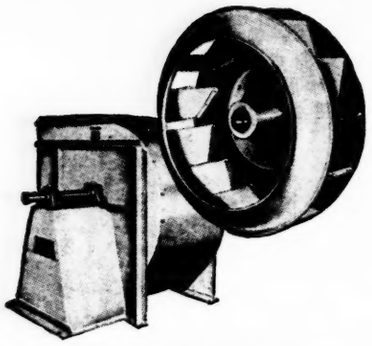

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POWER FIXED FANS
DESIGN 4
• AIR CONDITIONING •
HEATING &
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MASSACHUSETTS BLOWER DIVISION
The BISHOP & BABCOCK Mfg. Co.
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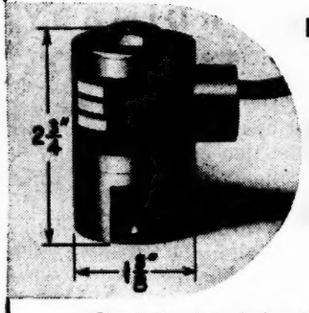
APPLICATIONS UNLIMITED
FOR ALLIED
Solenoid Valves



Check the data below to find where an Allied Control Three-Way, Two-Position Solenoid Valve will fit into your product design. Made of stainless steel, they safely and economically control or meter the flow of practically all fluid and gaseous media.

THREE STANDARD MODELS

Three way	two position
Two way	normally open
Two way	normally closed

Pressure range 5 to 250 psi
Dimensions 2¾" high x 1⅝" dia.
Impregnated coils withstand moisture and condensation
Soft insert seats prevent leakage
Spring loaded for positive action
Stainless steel parts highly resistant to corrosion and wear
Power consumption 10 watts max.
Pipe connections to specification
Models are approved by both Underwriter Laboratories and Factory Mutual
Also many special valves for handling pressures from 5 to 850 psi

For complete details and catalog write to

ALLIED CONTROL VALVE DIV.
The Skinner Chuck Co.
136 BELDEN AVENUE, NORWALK, CONN.

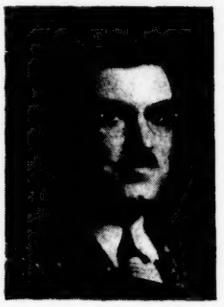
Perry M. Gish, assistant general traffic manager of the **American Radiator & Standard Sanitary Corp.**, has been promoted to general traffic manager to succeed Benjamin S. Thomas, who has retired.

Appointment of **J. Carl Bowen** as West Coast sales and service manager for the Heating and Air Conditioning Section of the Industrial Department, has been announced by the Machinery Division of **Dravo Corp.**, Pittsburgh.

Edward F. Lindsay has been appointed manager of the Ingersoll Utility Unit sales offices and display rooms in the Kimball Building, 306 South Wabash Ave., Chicago. **Ingersoll Steel Division, Borg-Warner**, has announced.

Getting Personal

Donald B. Robinson *(Condensation of Refrigerants on Finned Tubes*, page 83) was born in Calgary, Alberta, Canada. He received his B.A.Sc. from the University of British Columbia, and after a short time in the Royal Canadian Artillery, he returned to the university on a Standard Oil of California scholarship for post-graduate work. He received his M.A.Sc. in 1946. In the fall of 1946, Mr. Robinson accepted the Wolverine-Trufin Fellowship at the University of Michigan where he is now studying heat transfer and continuing his graduate work.



D. L. Katz

Donald L. Katz (co-author with Mr. Robinson of *Condensation of Refrigerants on Finned Tubes)* was born in Jackson County, Mich. In 1931 he received his B.S. in Chemical Engineering from the University of Michigan and his Ph.D. in 1933.

For three years he was employed by the Phillips Petroleum Co. of Bartlesville, Okla. to conduct research on oil and gas production. In 1936, Dr. Katz returned to the University of Michigan to teach chemical engineering. His researches in heat transfer, behavior of the hydrocarbons under pressure, and general engineering problems have resulted in over 60 technical papers. He has served many companies, including the Wolverine Tube Division, as consulting engineer.

He is a member of the ASME, American Chemical Society, American Institute of Mining Engineers, American Association for the Advancement of Science.

D. L. Fiske

David L. Fiske *(Humidity in Refrigeration*, page 63) is perhaps best known to readers as the editor of the several editions of The Refrigerating Data Book, published by the ASRE, of which he was formerly the secretary. The present article is the first of four papers drawn from a treatise by Mr. Fiske on the refrigeration low side. It is striking that much of the fundamental reasoning in refrigeration springs from humidity concepts long familiar in air conditioning.

This study was worked out in connection with investigations of the handling of perishable foods for the armed

forces overseas, taking the author, a Major in Quartermaster, to the Southwest Pacific in 1945. Subsequently he has been engaged in consulting engineering in New Rochelle, N. Y.

The author wishes to acknowledge the critical help of several engineers in his original paper and particularly that of Mr. D. D. Wile in connection with the four contributions to HEATING AND VENTILATING.

S. Oglesby, Jr.

Sabert Oglesby, Jr., *(Possibilities of Energy Storage in Heat Pump Applications,* page 69) received his B.S. from Alabama Polytechnic Institute in 1943. For the next three years he served as Lieutenant, U. S. Army Signal Corps. Upon return to civilian status he affiliated with Southern Research Institute, Birmingham, Ala., as research engineer. Mr. Oglesby receives his professional degree of electrical engineer from his alma mater in 1947.

Dr. E. N. Kemler, co-author with Mr. Oglesby, was head of the Engineering Research Division, Southern Research Institute. At present he is Assistant Director of Research, College of Engineering, New York University. His biography was published in June, 1945, page 116.

CONVENTION PROGRAMS

American Society of Refrigerating Engineers

A tentative program has been released of the papers to be presented at the 43rd annual meeting of the ASRE to be held at the Hotel Traymore, Atlantic City, N. J., December 7-10. The first day, Sunday, and Monday afternoon have been set aside for committee meetings. The convention will conclude with the induction of the new officers early Wednesday afternoon. The program follows:

Monday, December 8

10:00 a.m.—First Technical Session—
President Roland H. Money, presiding.

Graphical Aids to the Solution of Ammonia Absorption System Problems — A. B. Stickney, Armour & Co., Chicago, Ill.

Technical Factors in Greenhouse Air Conditioning—Dr. L. C. Marshall, University of California, Berkeley, Calif.

Modern Handling Methods in Refrigerated Storage—P. B. Christensen, Vice-President, and D. O. Haynes, Industrial Engineer, Merchants Refrigerating Co., New York, N. Y.

1:00 p.m.—Welcome Luncheon—
Speaker to be announced.

Tuesday, December 9

9:30 a.m.—Second Technical Session—
Clifford F. Holske, vice-president, ASRE, presiding.

Refrigeration Developments in the Dairy Industry—Dr. R. J. Ramsey, Ramsey Laboratories, Cleveland, Ohio.

Refrigeration Storage of Edible Nuts—C. T. Baker, Atlanta, Ga.

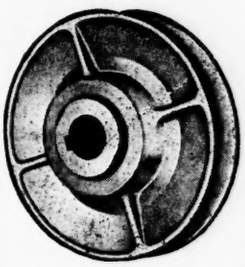
Mechanical Developments in Marine Refrigeration—S. W. Brown, Carrier Corp., New York, N. Y.

2:00 p.m.—Third Technical Session—
B. H. Jennings, vice-president, ASRE, presiding.

for **SMOOTH, EFFICIENT OPERATION....**

use **CENTRAL**

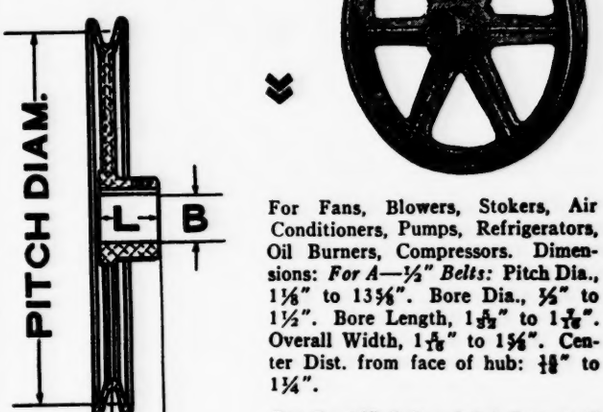
STANDARDIZED DRIVES



"CENTRAL" Variable Pitch Pulleys

Silent and true running, diamond bored to plus or minus .0005"—CENTRAL Variable Pitch Pulleys are available to meet all conditions for fractional horsepower drives on HEATING, VENTILATING, AIR CONDITIONING and similar equipment. Speed variations of approximately 30% may be obtained by including a Variable Pitch Pulley in the drive—variations of about 500% by using both pulleys of this type and by interchanging them on the motor and driven shafts. Grooves accurately built to the belt, assuring perfect grip. For following V-Belts: 0—3/8", A—1/2" and B—5/8". Dimensions: Bore diameter, 1/2" to 1". Width of belt groove, min. 1/2", max. 7/8". Overall length of pulley, 1 5/8".

"CENTRAL" V-Grooved Pulleys



For Fans, Blowers, Stokers, Air Conditioners, Pumps, Refrigerators, Oil Burners, Compressors. Dimensions: *For A—1/2" Belts:* Pitch Dia., 1 1/8" to 13 5/8". Bore Dia., 5/8" to 1 1/2". Bore Length, 1 3/32" to 1 7/16". Overall Width, 1 7/16" to 1 5/8". Center Dist. from face of hub: 11/16" to 1 1/4".

For B—5/8" Belts: Pitch Dia. 1 1/2" to 13 1/2". Bore Dia., 1/2" to 1 1/2". Bore Length, 1 1/4" to 1 13/32". Overall Width, 1 13/32" to 1 23/32". Center Dist. from face of hub, 11/16 to 1 1/4".

"CENTRAL" Flexible Couplings

Extreme simplicity of construction cuts assembling costs, eliminates constant adjusting and servicing. Consist of two metal flanges and one non-metallic insert. Available in wide variety of sizes. What are your transmission requirements?

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